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Heating · Piping and Air Conditioning

AUG 9 - 1929

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Journal Section
of A. S. H. V. E.

August, 1929

Heating·Piping
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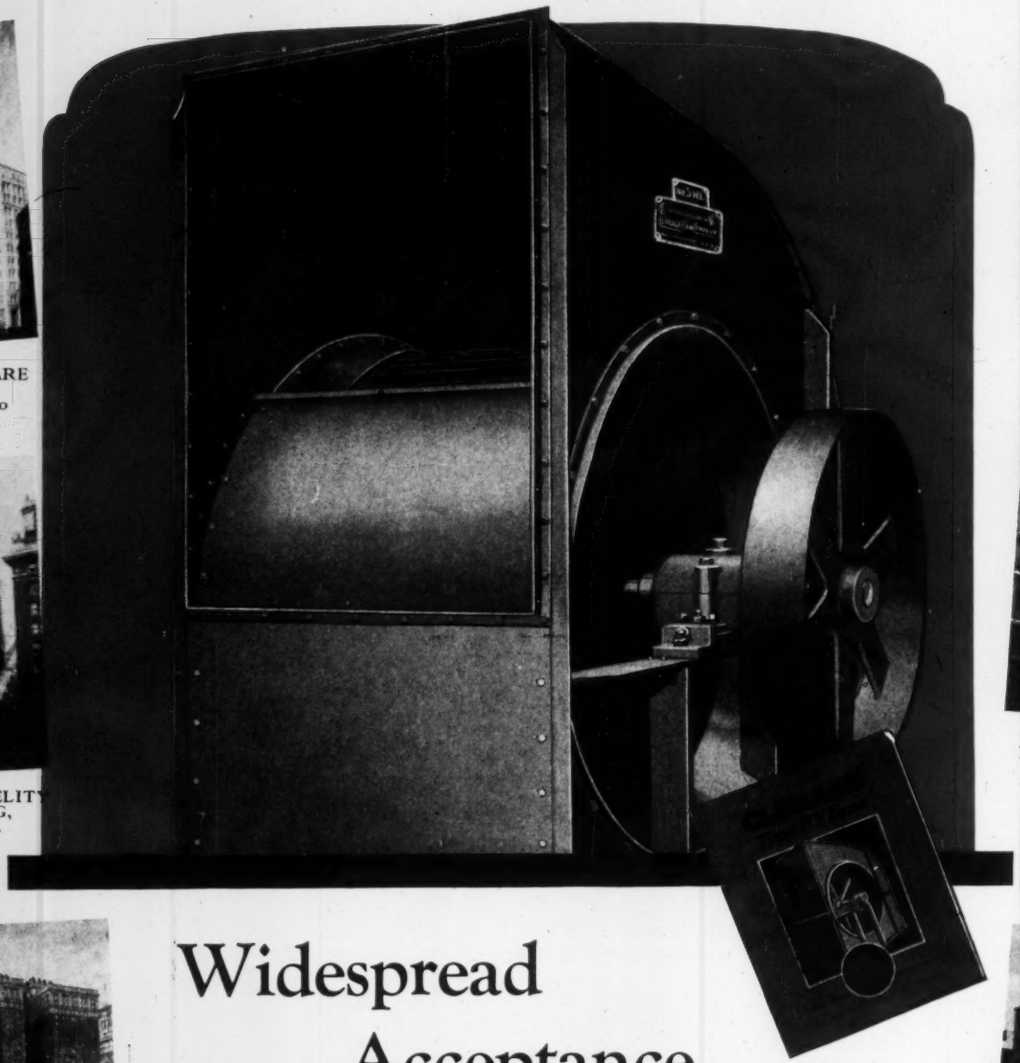

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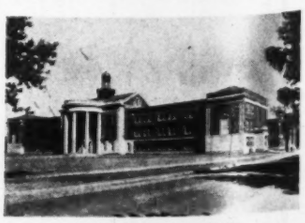

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The EDITOR'S PAGE

Heating · Piping and Air Conditioning

THE STAFF IS ENTHUSIASTIC ABOUT THIS ISSUE

When the editorial staff finished the preparation of this issue, it turned it over to the printing department and consigned it to its readers with the feeling that it represented the best combination of technical data and practical information yet presented within the covers of one number. Names like Giesecke, Woolrich, Kimball and Peebles, Crocker and Harman, Tomlinson and Hale mean a great deal in heating, piping and air conditioning, and there is an article from every one of them in this issue. These are backed up by many more articles on live, practical subjects and problems in this field—all building up a whole which we sincerely believe is of unequaled value.

We are enthusiastic about this issue to the point of risking the accusation of being egotistical. But it is the sort of enthusiasm, on the contrary, which is making it possible to enlist the interest and cooperation of leading engineers in producing the kind of articles this field needs and the kind the promise of which evoked such a fine response when HEATING, PIPING AND AIR CONDITIONING was first introduced to an immediately interested field.

HEAT TRANSMISSION THROUGH BUILDING

J. C. Peebles has, for twenty years, been a member of the faculty of Armour Institute of Technology. On the basis that research in this field should be co-ordinated in such a way that our readers will have knowledge of any developments in heating, piping or air conditioning here or abroad, we went to Professor Peebles with some Swedish data on heat transmission through building construction. We asked him to interpret it and present it to our readers. The first part of his interpretation starts with this issue.

EXACTING REQUIREMENTS IN AIR CONDITIONING

In air conditioning we lead with Malcolm C. W. Tomlinson and Leslie P. Hale.

Mr. Tomlinson started a thorough discussion of low humidities in last month's issue. His article then was on the subject of drying by low humidities. This month we present article 2 of his series. Number 2 is on the subject of human comfort at low humidities.

Leslie P. Hale describes for us the complete air conditioning installation in a modern cigar factory, in which the process of making cigars demands exact conditions of temperature and humidity at every stage. As Mr. Hale explains, the same reasons for which tobacco needs air conditioning apply to other hygroscopic materials. Therefore, a study of the method employed in a cigar factory results in an understanding which can be employed in many other industries.

TROUBLE EXPERIENCES IN HEATING

C. W. Kimball, member of the firm of R. D. Kimball Co., consulting engineers of Boston, and a member of our board of contributing editors, has prepared a most interesting article on how a new central heating plant was built around an old in Phillips Academy, Andover, Mass. This was quite an engineering feat, in that the more than forty-five buildings to be heated required uninterrupted service throughout the building of the new plant. Any engineer and contractor should find interesting information in such an article, which will appear in an early issue. In this number, Mr. Kimball presents some practical trouble experiences that he has encountered in heating. A pair of overalls in a steam line, for instance, can cause a lot of trouble.

PIPING—FLANGED JOINTS AND AUTHORITATIVE DESIGN DATA

As for piping, Sabin Crocker and John J. Harman have done a real job.

Sabin Crocker, who is an engineer in the engineering division of Detroit Edison Co., has prepared an article on "The Design of Flanged Pipe Joints" which contains so much valuable data that its presentation will require three issues. The first part appears in this number. His data represent absolutely new information in many cases, and we take pride in offering it as a real contribution to an understanding of the design of flanged pipe joints.

John J. Harman, research engineer of Walworth Co., Boston, Mass., has compiled and presented essential data having to do with the design, installation and maintenance of piping systems for semi-fluid or viscous liquids. Charts are offered in his article from which the engineer and contractor can work directly in the design of such piping systems. Industrial developments make the piping of viscous liquids highly important. Mr. Harman's article should, we feel, become the authority to which engineers will look on this subject.

Formulas from Giesecke

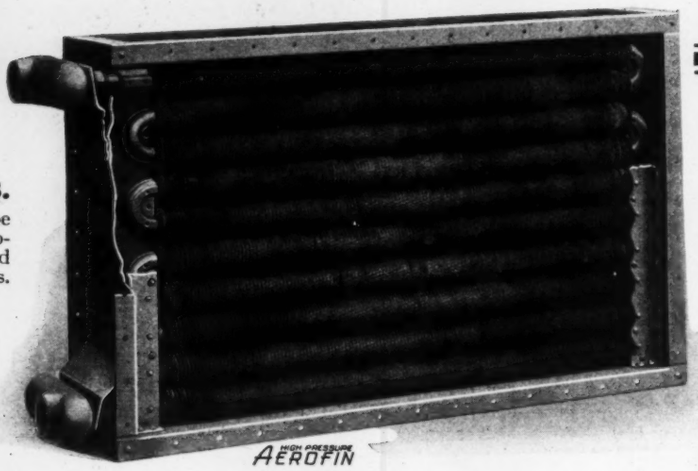
Professor Giesecke has made some thorough studies of "Cooling of Water in Iron Pipes," the subject of his article, and can present, therefore, formulas and data which the engineer and contractor can directly apply in the design and installation of heating systems. He not only presents such formulas and data, but he gives examples of their application. Nothing could be at once more technical and practical.

Heat and Energy

Professor Woolrich gives some fundamental definitions under the heading of "Heat and Energy Units," which are worth learning and applying.

Eliminating Fan Noise

There are many forms of noises made by fans, including air noise and machine noise. There are also several remedies, relating to placing of equipment and the construction details of the supports for fans. Cause and remedy are discussed in this issue and shown graphically by numerous sketches.



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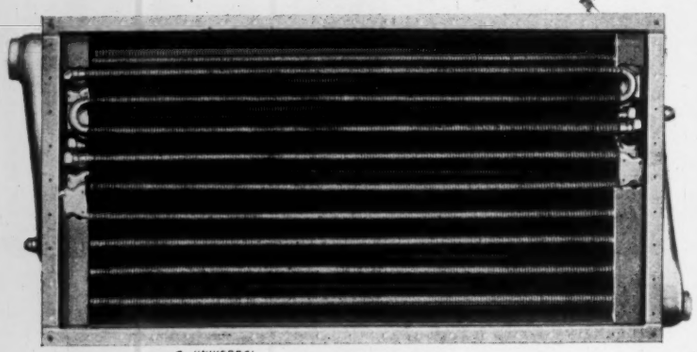

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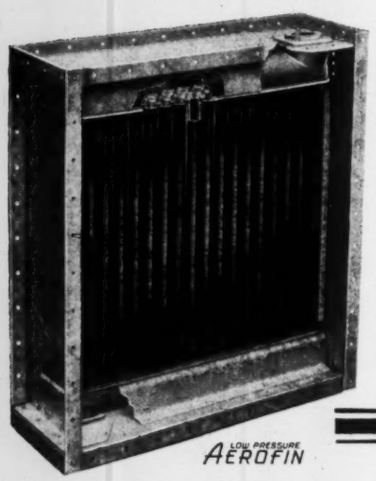

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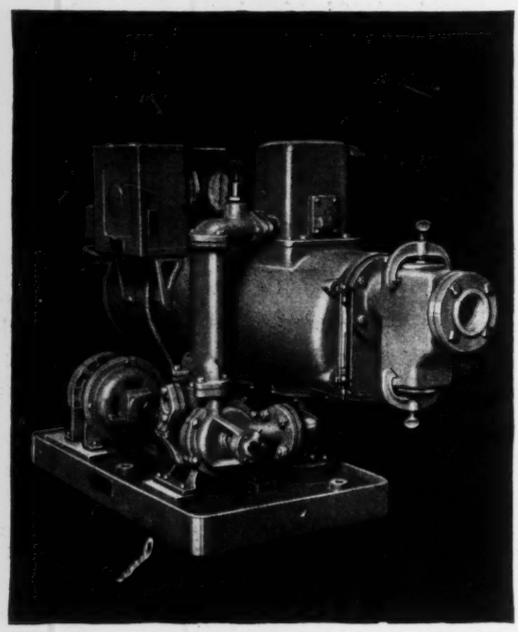

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Heating · Piping and Air Conditioning

Vol. I *August, 1929* **No. 4**

The Cooling *of* Water *in* Iron Pipes

By F. E. Giesecke, Ph. D.*

WHEN water flows through pipes, surrounded by air of a lower temperature than that of the water, heat will be dissipated by the pipe and the water will be cooled.

The dissipation of heat by such a pipe is partly by conduction of the surrounding air, partly by convection by that air, and partly by radiation from the iron pipe.

Since air is a poor conductor of heat, the portion of the total dissipated heat removed from the pipe by conduction into the surrounding air is so small that it may generally be neglected.

The quantity of heat removed from the pipe by convection currents in the surrounding air varies with the relative temperatures of the air and the pipe and with the velocity of the air current.

In the following discussion it is assumed that air is normally

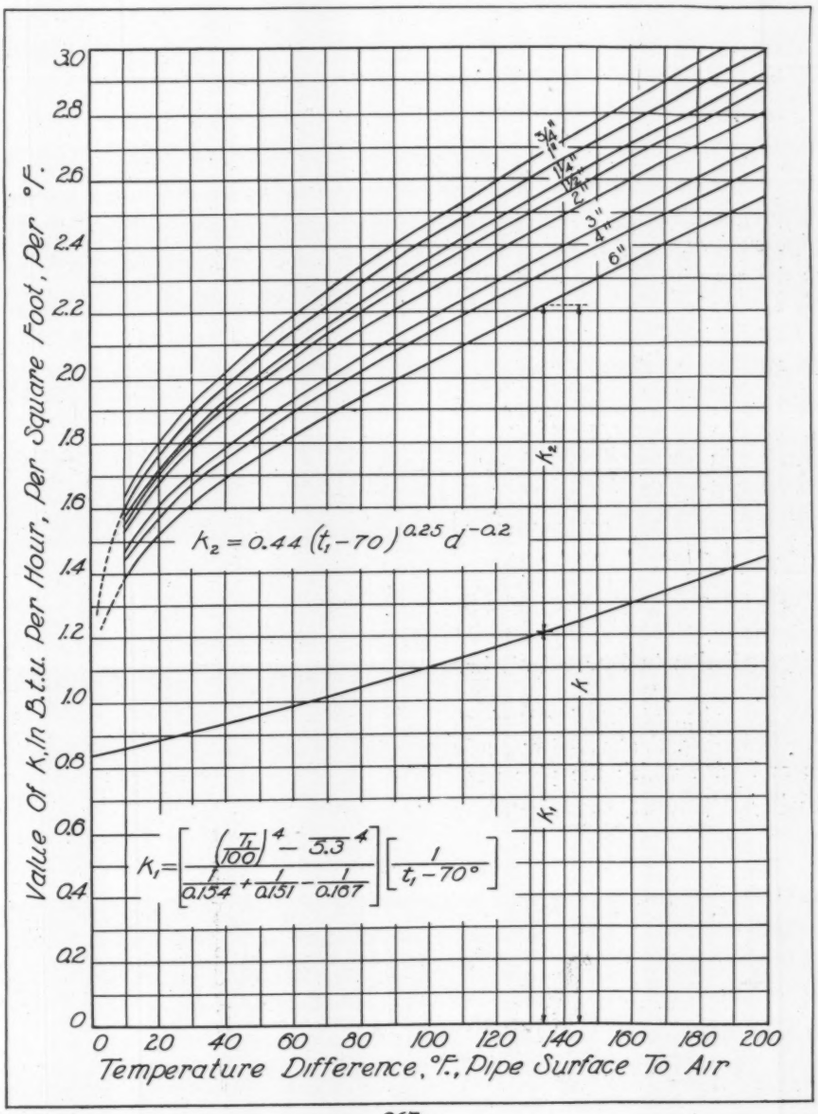


FIG. 1

*Director, Texas Engineering Experiment Station, College Station, Texas.

at rest and that the convection currents which remove heat from the pipe are induced by the temperature difference, pipe to air.

The quantity of heat radiated by the pipe to the surrounding objects in excess of the heat radiated by the surrounding objects to the pipe varies with the difference of the fourth powers of the absolute temperatures of the pipe and of the surrounding objects.

In calculating the heat lost by radiation from a pipe like that under discussion, it is generally assumed that the temperatures of the surrounding objects are the same as that of the surrounding air. This assumption is frequently inaccurate. For example, when a single pipe line is employed to heat a room, the surrounding objects may be the floor, ceiling and walls, and the temperatures of these surfaces may differ materially from each other and also from that of the air surrounding the pipe. When a set of pipes is used to heat a room, a portion of the objects surrounding the pipe are the other pipes; the surface temperatures of those pipes differ materially from temperature of surrounding air.

The dissipation of heat by iron pipes has been studied and discussed by Dr. F. Wamsler, Dr. H. Rietschel, Professor A. H. Barker, Dr. W. Nusselt, L. B. McMillan, R. H. Heilman, by the writer, and by many others. After a careful study of the data secured by these investigators, as well as of the conclusions arrived at by them, the writer recommends the following two formulas as giving results which are quite accurate, and certainly sufficiently accurate for all problems arising in engineering practice:

Formula (1) gives the quantity of heat dissipated by radiation from the pipe surface.

Formula (2) gives the quantity of heat dissipated by air conduction and air convection.

The sum of the two quantities of heat is the total heat dissipated by the pipe surface.

Formula (1) is

$$K_1 = \frac{\left(\frac{T_1}{100}\right)^4 - \left(\frac{T_2}{100}\right)^4}{\frac{1}{c_1} + \frac{1}{c_2} - \frac{1}{c}} \quad \frac{1}{t_1 - t_2}$$

where

K_1=heat dissipated by radiation in B.t.u., ft.$^{-2}$, hr. $^{-1}$, F. $^{-1}$;

T_1=temperature, in deg. F. abs., of the pipe surface;

T_2=average temperature, in deg. F. abs., of the surfaces surrounding the pipe and with which the pipe surface exchanges radiant energy;

C_1=radiation factor of the pipe surface;

C_2=radiation factor of the surfaces surrounding the pipe;

C=radiation factor of the black body.

To apply this formula, the values of C may be taken from Table I. The formula was first proposed by Doctor Wamsler, and is based on the reasoning presented in the concluding section at the end of this article.

Formula (2) is:

$$K_2 = 0.44\ (t_1, -t_2)^{0.25}\ d^{-0.2}$$

Where K^2 = heat dissipated by air conduction and air convection in B.t.u. ft. $^{-2}$, hr. $^{-1}$ deg. F.$^{-1}$

t_1 = temperature of pipe surface in deg. fahr.

t_2 = average temperature of air surrounding the pipe in deg. fahr.

d = outside diameter of pipe in inches.

This formula was proposed by Doctor Wamsler, modified by Doctor W. Nusselt, and then further modified by the writer so that the results determined by means of the formula agree well with the experimental results

Table I

RADIATION FACTORS
TRANSLATED FROM HUTTE

Black body*	0.167
Lamp black	0.154
Glass	0.154
Iron (rough & oxidized)	0.154
Mortar	0.150
Water	0.112
Ice	0.107
Sandstone	0.100
Granite	0.075
Earth	0.063
Iron (highly polished)	0.050
Gravel	0.048
Copper (slightly polished)	0.028

*The "black body" listed here is a purely imaginary one that would absorb all radiant energy impinging upon it. No such body exists in nature.

secured by L. B. McMillan, (Transactions A. S. M. E. Vol. 37, p. 941), as well as with those secured by R. H. Heilman (Transactions A. S. M. E. Vol. 44, p. 303), and with those secured by the writer, (Thesis in Graduate School, University of Illinois, 1924).

From these two formulas the diagrams of Figs. 1 and 2 were constructed. These diagrams can be used to solve the following types of problems:

Problem 1. *A room is to be supplied with* 9000 *B.t.u. per hr. and to be heated to* 70 *deg. with a* 2*-in. pipe, containing steam at a pressure of* 1 *lb. How many feet of* 2*-in. pipe should be used?*

From a steam table we find that the temperature of steam of 1 lb. pressure is 215 deg. Since iron is a good conductor of heat, we will assume that the temperature of the outer surface of the bare pipe is only about 2 per cent less than that of the steam, or that it is 211 deg. The temperature difference, pipe surface to air, will then be 211-70, or, 141 deg.

Referring to Fig. 1, we find 140 on the lower margin of the diagram, estimate where 141 would be, and move from that point in a vertical direction to the intersection with the 2-in. curve; from this point of intersection, move horizontally to the left margin of the diagram, and find the value of k to be 2.5. This means that every square foot of the external surface of the pipe will dissipate 2.5 B.t.u. per hour and per degree temperature difference. Since the temperature difference is 141 deg. every square foot will dissipate 2.5 × 141 or 353 B.t.u. Since the room requires 9000 B.t.u. per hour, we need

9000/353, or 25.5 square feet. From Table II we find that, for a 2-in. pipe, 1.61 feet are needed to supply 1 square foot of external surface; hence, we need 25.5 × 1.161, or 41 feet of 2-in. pipe.

Problem 2. How many feet of 1¼-in. pipe should be used?

Proceed as explained for Problem 1, and find that 55 feet of 1¼-in. pipe should be used.

Problem 3. If 2-in. pipe is used with water flowing through it at an average temperature of 200 deg., how many feet of pipe are needed?

As suggested in Problem 1, we assume the temperature of the pipe surface to be 196 deg.; the temperature difference, pipe surface to air, will then be 196-70, or, 126 deg. For this temperature difference and a 2-in. pipe, we find, from Fig. 1, the value of k to be about 2.41. The heat dissipated by the pipe will be 2.41 × 125, or 304 B.t.u. per square foot, per hour. Since 9000 B.t.u. are to be supplied, we will need 9000/304, or 29.6 square feet, and 29.6 × 1.61, or 48 ft. of 2-in. pipe.

Problem 4. How much, in the preceding problem, will the water cool in flowing through the pipe, if its average velocity is three feet per second?

The velocity of the water is 3 ft. per second or 10,800 ft. per hr. The total volume of water flowing through the pipe per hour is 10,800/42.9, or 264 cu. ft. This is equal to 264 × 60.12, or 15,870 pounds. Since this quantity of water dissipates 9000 B.t.u. per hour, it will cool 9000/15870, or 0.57 degrees.

Table 2

STANDARD WROUGHT IRON PIPES

NOMINAL INTERNAL DIAMETER, INCHES	ACTUAL EXTERNAL DIAMETER, INCHES	ACTUAL INTERNAL DIAMETER, INCHES	LENGTH OF PIPE WEIGHING 1 LB., FEET	LENGTH OF PIPE HAVING 1 SQ. FT. OF EXTERNAL SURFACE, FEET	LENGTH OF PIPE CONTAINING 1 CU. FT. FEET
½	0.84	0.62	1.118	4.55	472.4
¾	1.05	0.82	0.882	3.64	270.0
1	1.32	1.05	0.594	2.90	166.9
1¼	1.66	1.38	0.437	2.30	96.3
1½	1.90	1.61	0.366	2.01	70.7
2	2.38	2.07	0.272	1.61	42.9
2½	2.88	2.47	0.172	1.33	30.1
3	3.50	3.07	0.131	1.09	19.5
3½	4.00	3.55	0.109	0.95	14.6
4	4.50	4.03	0.092	0.85	11.3
4½	5.00	4.51	0.079	0.76	9.0
5	5.56	5.05	0.067	0.69	7.2
6	6.63	6.07	0.058	0.58	5.0
8	8.63	7.98	0.035	0.44	2.9
10	10.75	10.02	0.024	0.36	1.8
12	12.75	12.00	0.020	0.30	1.3

Problems similar to Problem 4 arise frequently in heating practice and can be solved as shown for this problem. However, the solutions can be simplified materially by the use of the diagram of Fig. 2. This diagram was prepared from that of Fig 1, and shows the cooling of water in bare iron pipes located in 70 deg. air when the length of the pipe is one foot and the velocity of the water is one inch per second.

To apply the diagram to the solution of Problem 4, assume, as above, that the temperature of the pipe surface is 196 deg.; the temperature difference, pipe surface to air, will be 196-70, or 126 deg. Find 120 on the lower margin of the diagram and estimate the location of 126; from this point move vertically to the intersection with the 2-in. curve and from there move horizontally to the left margin of the diagram and read, approximately, 4.3. Dividing this by 10, as directed by the note on the left margin, gives 0.43 deg. cooling, if the pipe were one foot long and the velocity one inch per second. Since the pipe is 48 feet long, the cooling is 48 times as great, and since the velocity is 36 inches per second, the cooling is 36 times as small.

The cooling is, therefore:

0.43 × 48/36, or 0.57 deg. as calculated above.

Problem 5. The main of a one-pipe heating system is a 3-in. bare pipe, 150 ft. long; the average temperature of the water in the pipe is 190 deg.; its velocity is 2 in. per sec.; the temperature of the surrounding air is 60 deg. How many degrees will the water cool in flowing the distance of 150 ft.?

Use the diagram of Fig. 2 as follows: Assuming the pipe surface temperature to be 186 deg., the temperature difference will be 186-60, or 126 deg. Find 126 on the lower margin, move vertically to the 3-in. curve and thence horizontally to the left margin and read 2.75. The cooling will be 0.275 × 150/2 or 21 deg.

Problem 6. If, in Problem 5, the pipe is covered with pipe covering having an efficiency of 80 per cent, what will be the cooling?

If a pipe covering has an efficiency of 80 per cent, it permits a heat loss of only 20 per cent of that of the bare pipe. In this case, the heat loss would, therefore, be 20 per cent of 21 deg. or 4 deg.

Problem 7. A second floor radiator is supplied with 500 lb. of water per hr. at a temperature of 190 deg.; the room temperature is 70 deg. The riser is 12 ft. long; how many degrees will the water cool if the riser is a ¾-in. pipe, and how many degrees, if it is a 1-in. pipe?

As the unit weight of 190 deg. water is 60.36 pounds, 500 pounds are equivalent to 8.3 cu. ft. The velocity of the water will, therefore, be 8.3 × 270 × 12/3600, or 7.5 inches per second in the ¾-in. pipe and 4.6 inches per second in the 1-in. pipe.

Using the diagram, as explained above, we find the cooling to be 1.9 deg. in the ¾-in., and 2.3 deg. in the 1-in. pipe.

This calculation shows that when the flow and return risers of a radiator are of different sizes it is best to use the smaller pipe for the flow riser.

Problem 8. At what rate does the water cool in flowing through the pipe lines of a gravity-circulation greenhouse heating system?

Let the system be constructed as shown in Fig. 3. Assume that the temperature of the air in the greenhouse is 60 deg. and that the water reaches the upper end of the flow riser at a temperature of 180 deg., when the system is operating at a uniform rate. As the water flows through the system, the cooling takes place partly in the upper flow main, partly in the return risers leading to the heating coils, partly in the heating coils, and partly in the return line connecting the heating coils to the heater. (The flow riser should be covered and the cooling in it may be neglected.)

The difference in the weight of the water in the flow

and return risers produces the pressure head which maintains circulation and which is equal to the friction head in the entire system during the period of uniform operation. During that period, the water will flow with a uniform velocity through the system. What that velocity is, we do not know, but we may determine it by trial calculations.

Let us begin by assuming that the velocity in the 4-in. flow main is 3 inches per second. In that event, the velocities in the other pipes will be as follows:

In the two 3-in. lines: $3\times19.5/2\times11.3$, or 2.6 inches; in the eight 2-in. lines, $3\times42.9/8\times11.3$, or 1.4 inches.

To find the cooling in the flow main, assume that it is about 10 deg., so that the average temperature of the water in the flow main will be about 175 deg. The temperature difference, pipe surface to air, will then be about 115 deg., and the cooling in the flow main (120 feet long) will be, from Fig. 2, $0.18\times120/3$, or 7 degrees, and the water will enter the return riser at 173 deg.

The cooling in the 3-in. return lines will be, from Fig. 2, $0.24\times9/2.6$, or 1 degree, and the water will enter the heating coils at 172 deg.

The cooling in each of the eight 2-inch lines will be, from Fig. 2, 0.34x120/1.4, or 29 degrees, and the water will enter the return connections at 143 deg.

The over-all cooling in the system will be 37 degrees.

For a velocity of 3 inches per second in the 4-in. main, the quantity of water flowing through the system is

$3\times3600\times60.6/12\times11.3$, or 4820 lb. per hr., and the total heat output is 4820x37, or 178,000 B.t.u. per hr.

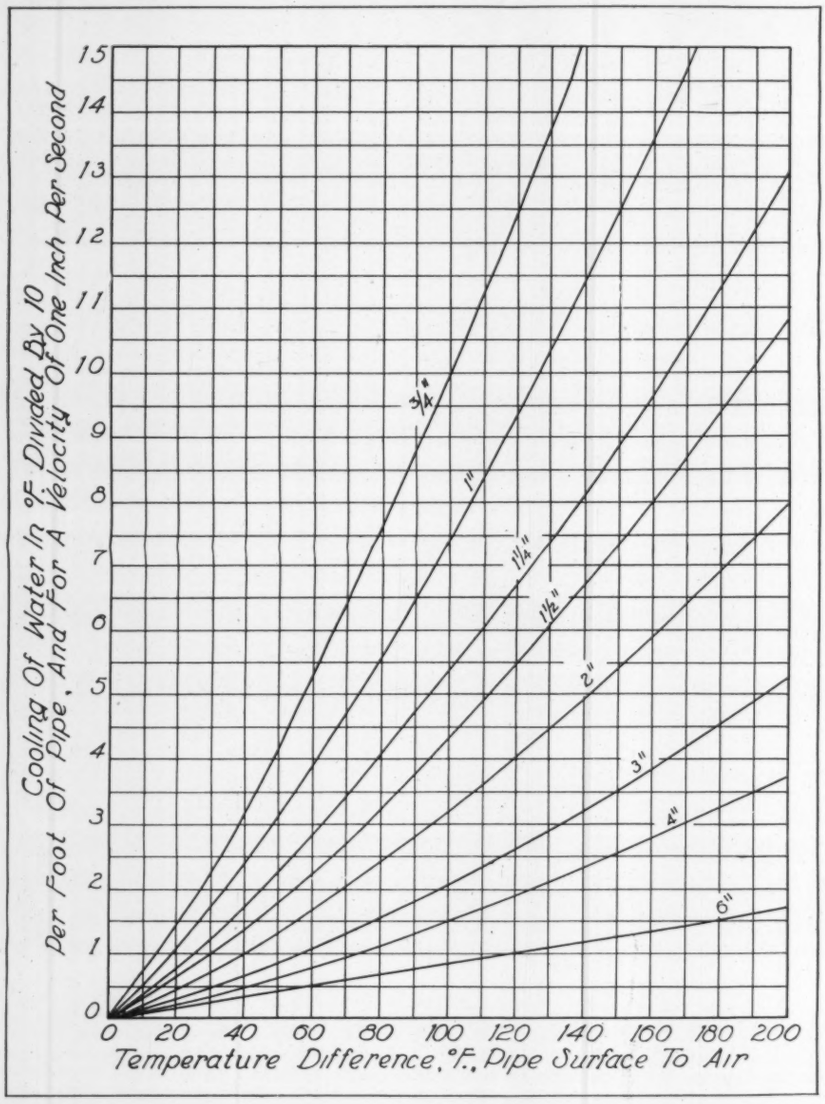


FIG. 2

By using the method described in the A. S. H. V. E. GUIDE and also in "Design of Gravity Circulation Water Heating System" (Giesecke) we find that, for the assumed circulation through the system, the pressure head would be 560 milinches and the friction head, 290 milinches. This proves that the assumed velocity of 3 inches per second was too low.

By repeating the calculation and assuming a velocity of 4 inches in the flow main, we find a total temperature drop of 26 degrees, a total heat output of 166,000

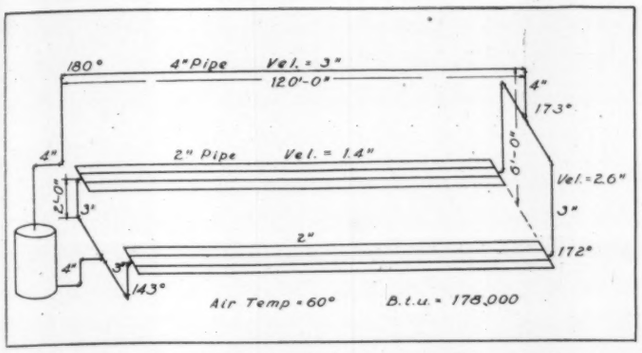


FIG. 3—COOLING OF WATER IN GREENHOUSE HEATING SYSTEM

B.t.u. per hr., a pressure head of 370 milinches, and a friction head of 480 milinches. This proves that the assumed velocity of 4 inches is too high.

We may conclude, therefore, that the actual velocity in the flow main will be between 3 and 4 inches per second and the actual heat output between 178,000 and 166,000 B.t.u. per hr.

The calculation described above is sufficiently accurate for practical purposes, but not absolutely accurate. For instance, it was assumed in the calculation that the 4-inch main is surrounded by bodies having a temperature of 60 deg., whereas, the glass roof and walls will be colder than 60 deg. when the greenhouse is losing heat at the rate of about 170,000 B.t.u. per hr. Consequently, the 4-in. main will dissipate more heat than calculated. On the other hand, the 2-in. pipes of the heating coils will dissipate less than the quantity calculated, because they are surrounded by objects having a higher temperature than 60 deg., when the system is in operation as described.

Formula 1, presented at the start of this article, was, as explained, proposed by Doctor Wamsler and based on the following reasoning:

Every object radiates heat to its surroundings and receives heat from them.

The quantity of heat radiated by an object varies with the character of its surface and is proportional to the fourth power of the absolute temperature of that surface.

Hence, we have

$$Q = A\,C\left(\frac{T}{100}\right)^4 = A\,E \ldots\ldots\ldots (3)$$

Where

Q = heat, in B.t.u., radiated per hour by a given object;

A = area of the surface of the object, in square feet;

C = radiation factor (multiplied by 100^4) of the surface, in B.t.u., per hr., per sq. ft. The radiation factor is equal to the number of B.t.u. radiated by the surface per hr. and per sq. ft. when the temperature of the surface is one degree absolute. The factor 100^4 is introduced to simplify later calculations.

T = absolute temperature of the surface in deg. Fahr.

E = heat emission of the particular surface, in B.t.u. per hr., per sq. ft., for the temperature T.

Let B_1 and B_2 represent small portions, say one square foot, of infinitely large parallel plane bodies, and let each of the bodies be protected by a perfect mirror, M, so that no heat rays can pass through either body and so that all heat rays passing from one of the bodies to the other will be either absorbed or reflected by that body.

In considering this to-and-fro reflection, the reader should bear in mind that its velocity is the same as that of light, namely, about 186,000 miles per second. Also, that there is no difference between the two types of radiation known, respectively, as heat and as light, except wave length. These wave lengths are extremely short. They are measured in angstroms. Ten million angstroms are equal to one millimeter. The wave length which the human eye can sense, and which we call light, varies from about 7,000 angstroms for red light to about 4,000 angstroms for violet light. Radiations of longer wave length than about 7,000 angstroms are called infra red. They include the heat radiation of the ordinary heating coils and radiators and the long waves used in wireless circuits. Radiations of shorter wave length than about 4,000 angstroms are called ultra violet; they include the Roentgen rays which may have wave lengths as short as one angstrom.

The total heat (B.t.u.) radiated by the body, B_1, per square foot and per hour is its emissivity and is represented by E_1. (At 180 deg. fahr., the emissivity of rough cast iron is about 258 B.t.u., and that of polished copper, about 84 B.t.u.).

The fractional part of the total heat radiated upon the body B_1 which is absorbed by A_1 is the absorbivity of B_1 and is represented by a_1. (The absorbivity of rough cast iron is about 92 per cent and that of polished copper is about 30 per cent.).

The corresponding properties of the body B_2 are represented, respectively, by E_2 and a_2.

Considering the reciprocal heat exchange between the bodies B_1 and B_2, we note that B_1 radiates E_1 B.t.u. to B_2; of this quantity, B_2 absorbs $E_1\,a_2$ and reflects $E_1\,(1-a_2)$ back to B_1 which, in turn, absorbs $E_1\,(1-a_2)\,a_1$ and reflects $E_1\,(1-a_2)\,(1-a_1)$, and so on, indefinitely.

Considering the quantity of heat E_1 radiated by B_1 toward B_2, and representing $(1-a_2)\,(1-a_1)$ by k, we note that $E_1\,a_2\,(1+k+k^2+k^3+\text{etc.})$, or $\frac{E_1\,a_2}{1-k}$ B.t.u. are absorbed by B_2;

and that

$E_1\,(1-a_2)\,a_1\,(1+k+k^2+k^3+\text{etc.})$ or $\frac{E_1\,(1-a_2)\,a_1}{1-k}$

B.t.u. are reflected back to and reabsorbed by B_1.

Similarly, of the quantity of heat E_2, radiated by B_2 toward B_1, the body B_1 absorbs $\frac{E_2\,a_1}{1-k}$ B.t.u.

When the reciprocal heat radiation assumes a steady state, the two bodies will be of like temperature. At that time, the total heat radiated by B_1 is equal to the total heat absorbed by B_1; i. e.,

$$E_1 = \frac{E_1\,(1-a_2)\,a_1}{1-k} + \frac{E_2\,a_1}{1-k} \quad \text{and, hence,} \quad \frac{E_1}{a_1} = \frac{E_2}{a_2}$$

To facilitate computations, the concept of a "black body" was introduced by Kirchhoff. A black body is

one which absorbs all heat radiation impinging upon it. There is no black body in nature. The nearest approach to a black body in nature is a small aperture leading into a large cavity. All heat radiation passing through such an aperture enters the cavity and is there reflected to and fro, so that practically no part thereof escapes outward through the aperture. Lamp black has an absorption of about 92 per cent of that of the black body.

If we represent the emissivity and absorbivity of the black body, respectively, by E and a, and remember that the black body absorbs all heat radiated upon it and that, therefore, the value of a is 100 per cent, or 1, we have: $\frac{E_1}{a_1} = \frac{E_2}{a_2} = E$, or, in words:

The ratio of heat emissivity to heat absorbivity is the same for all bodies and is equal to the heat emissivity of the black body. This relation was first established by Kirchhoff, and is known as Kirchhoff's law. Further analysis shows that the law is strictly true only for radiation of the same wave length and for a fixed temperature. For the purpose of calculations relating to heating systems, it may be considered true for all wave lengths and all temperatures.

If B_1 is at a higher temperature than B_2, the quantity of heat lost by B_1 to B_2 is equal to the total heat radiated by B_1 minus the total heat absorbed by B_1, or equal to

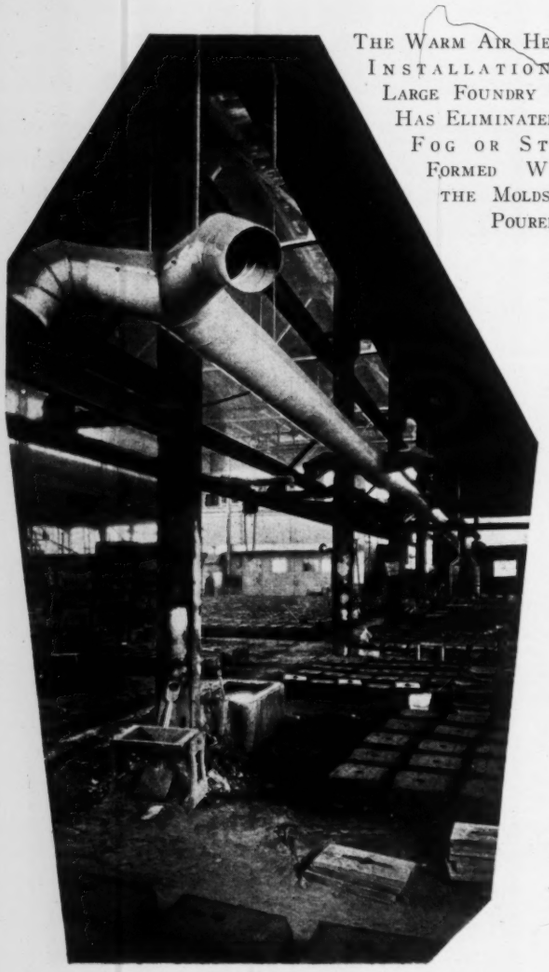
$$Q = E_1 - \frac{E_1 (1-a_2) a_1}{1-k} - \frac{E_2 a_1}{1-k}$$ B.t.u. per hr. per sq. ft.

By substituting in this equation the following values
$E_1 = C_1 (T_1/100)^4$; $E_2 = C_2 (T_2/100)^4$
$k = (1-a_1)(1-a_2)$
$a_1 = C_1/C$ and $a_2 = C_2/C$ because
$E_1/a_1 = E. \therefore a_1 = E_1/E = C_1 (T_1/100)^4 \div C (T_1/100)^4 = C_1/C$, where C is the radiation coefficient of the black body, the equation will reduce to:

$$Q = \frac{\left(\frac{T_1}{100}\right)^4 - \left(\frac{T_2}{100}\right)^4}{\frac{1}{c_1} + \frac{1}{c_2} - \frac{1}{c}}$$ B.t.u. per hr. per sq. ft.

transferred by radiation from the body B_1 to the parallel and adjacent body B_2.

Further analysis shows that the same heat transfer takes place when the body B_1 is completely surrounded by the body B_2 and that the heat transfer is not affected by the distance between the two bodies. The formula may therefore be used to calculate the heat losses by radiation of a single pipe surrounded by objects for which an average temperature (T_2) and an average coefficient of radiation (C_2) may be assumed.



The Warm Air Heating Installation in a Large Foundry That Has Eliminated the Fog or Steam Formed When the Molds Are Poured

Warm Air Heating in a Foundry

THE illustration presented here shows the application of warm air heating in one of the large foundries of the Forest City—Walworth Run Foundry Co., Cleveland, Ohio. This is the foundry at West 49th street and Maywood avenue. As designed, the heating plant has about one-third to one-fourth of the capacity of the building in which it is located. The foundry is 300 feet long by 150 feet wide and has a great expanse of glass in the side walls. The peak of the monitor is 36 feet high.

It is claimed that one of the problems solved by this heating system is the elimination of the annoying formation of steam and fog at the time of pouring the melted metal or when the outdoor humidity is high. The fog or steam usually forms when the molds are being poured and frequently offers serious hazards in connection with handling the hot metal. With the new heating plant in operation, it is claimed that this trouble is entirely eliminated in that portion of the foundry most vitally concerned.

This heating system, the character of which may be seen in the illustration, is equipped with a furnace having a 32-in. fire pot with a 60-in. casing. It is an exceptionally heavy model, the casting alone weighing about 3,000 pounds. To offset the static pressure built up in the duct, a blower directly connected has been installed on the return side of the system. Note in the illustration the length of the conductor pipe.

Piping Viscous Liquids

By J. J. Harman

PIPE lines for transporting materials of many different kinds are being employed throughout the civilized world today a great deal more extensively than ever before, and evidences of the increasing utilization of the great economies to be gained by this method of transportation are apparent on every hand.

A striking illustration of this may be found in the fact that the map of the United States is becoming liberally striped with trunk pipe lines for transporting oil, rivaling in extent our vast system of railroad transportation. Gas and water are also carried long distances across country by large pipe lines and distributed through systems of smaller pipes to the many places where they are used at the centers of population and industry.

We seldom stop to consider in the daily routine of this modern age how dependent we are on piping systems for the common necessities of our everyday life. All the materials that contribute to our welfare exist in one or more of the three general states, solid, liquid or gaseous, and of these, gases and liquids can be readily carried in pipes, while solids also can be successfully handled when pulverized and carried in suspension in a liquid or gaseous fluid. In present day commercial practice, steam, water, air, oil, gas, process liquids, semi-fluid liquids, and solids in suspension are all successfully handled in pipe lines.

As indicated by the title, it is the purpose of this paper to discuss some of the special problems encountered in the design, installation and maintenance of piping systems for viscous or semi-fluid liquids which lie in the border land between the liquid and the solid states. Conspicuous examples of this class of adhesive, slow-

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liquids which lie in the border land between the liquid and the solid states. Conspicuous examples of this class of adhesive, slow-moving liquids are animal, vegetable and mineral oils, bituminous compounds, syrups, chemicals and process liquids.

Description of Oil	Before Cracking		After Cracking	
	Gravity Baume' °	70° Viscosity Saybolt	Gravity Baume' °	70° Viscosity Saybolt
Heavy Mexican Flux Oil	10.8	14,500	12.6	530
California Crude	16.9	5,400	15.5	414
Heavy Mid Continent Fuel	23.5	810	21.2	135
Garlex Okla. Fuel	31.3	183	28.0	70

FIG. 1.—GRAVITY AND VISCOSITY OF OILS BEFORE AND AFTER CRACKING

Description of Liquid		Col. 1 Temp. 100° F	Col. 2 Temp 212° F	Ratio Col. 2 : Col. 1
Water		.00684	.00284	.415
Lubricating Oil	A	.7870	.1231	.156
Lubricating Oil	B	1.6696	.1780	.107
Lubricating Oil	C	2.6385	.2490	.094
Lubricating Oil	D	7.2595	.4534	.062
Lubricating Oil	E	16.4998	.6540	.040

$$\text{Kinematic Viscosity} = \frac{\text{Absolute Viscosity in Poises}}{\text{Density in grams per cubic centimeter}}$$

FIG. 2.—KINEMATIC VISCOSITY OF WATER AND FIVE LUBRICATING OILS

FIG. 3.—CONVERSION TABLE FOR REDWOOD, SAYBOLT AND ENGLER VISCOSIMETERS

Time, Engler	Time, ratio			Time, Redwood	Time, Engler	Time, ratio			Time Redwood
	Engler-Saybolt	Engler-Redwood	Saybolt-Redwood[a]			Engler-Saybolt	Engler-Redwood	Saybolt-Redwood[a]	
Sec.				Sec.	Sec.				Sec.
56	1.73	2.61	1.51	21.5	140	1.55	1.81	1.17	77.5
58	1.72	1.93	1.12	30.0	150	1.54	1.80	1.17	83.2
60	1.71	1.93	1.13	31.2	160	1.53	1.80	1.17	89.1
62	1.70	1.92	1.13	32.2	180	1.53	1.80	1.17	100.3
64	1.69	1.91	1.13	33.4	200	1.52	1.79	1.18	111.9
66	1.68	1.91	1.13	34.6	225	1.52	1.79	1.18	126.0
68	1.68	1.90	1.14	35.8	250	1.51	1.78	1.18	140.3
70	1.67	1.90	1.14	36.9	275	1.51	1.78	1.18	154.4
75	1.65	1.88	1.14	39.8	300	1.51	1.78	1.18	168.5
80	1.63	1.87	1.15	42.7	325	1.51	1.78	1.18	183.0
85	1.62	1.86	1.15	45.7	350	1.50	1.78	1.18	197.0
90	1.61	1.86	1.15	48.5	375	1.50	1.77	1.18	211.3
95	1.60	1.85	1.16	51.4	400	1.50	1.77	1.18	225.5
100	1.59	1.84	1.16	54.3	500	1.50	1.77	1.18	282.0
110	1.58	1.83	1.16	60.1	600	1.50	1.77	1.18	339.0
120	1.56	1.82	1.16	66.0					
130	1.56	1.81	1.17	71.7					

[a] These values were calculated directly from the times of discharge and differ in some cases by 0.01 from values calculated from the two previous columns.

moving liquids are animal, vegetable and mineral oils, bituminous compounds, syrups, chemicals, and process liquids of various kinds.

High Viscosity Increases Pressure Drop

In piping liquids of this character which may have viscosities several hundred times greater than that of water, it is evident that larger pipe sizes must be used than are necessary for piping water, because of the greatly augmented frictional resistance due to the increased "skin friction" of the viscous fluid flowing against the inner wall of the pipe and also to the greater "molecular internal friction" of the liquid itself.

In spite of this obvious fact, the writer has in mind at the present moment an actual piping installation for fuel oil which was laid out in accordance with the ordinary practice for water pipes without taking into consideration the relatively high viscosity of the oil, and when the piping system was placed in operation the unsatisfactory performance came both as a disappointment and a surprise. An oversight of this kind is perhaps uncommon but even when the sluggish character of the liquid is realized, the determination of the proper size of pipe to use for a definite set of conditions is not an easy problem to solve, because it involves a number of different factors, and there are numerous empirical formulas in more or less common use which give results that are far from correct if they are applied to conditions different from those for which the formulas were developed.

It is a rather common fallacy to consider the specific gravity of an oil a measure of its flow characteristics, but this is only true in a rough way for oils of similar composition. In general the specific gravity does not determine the viscosity as is clearly indicated by the data shown in Fig. 1.

Temperature Affects Viscosity

The pressure drop caused by the frictional resistance to flow in a pipe line depends to a large extent on the degree of mobility possessed by the liquid, but in the case of semi-fluid liquids it is also greatly influenced by temperature, because, unlike the viscosity of water which is affected comparatively little by changes in temperature, the viscosity of a semi-fluid liquid often varies through a wide range with a comparatively small rise or fall in temperature. It is necessary, therefore, to know the viscosity throughout the temperature range anticipated in service.

The change in viscosity of oil at temperatures above the wax separation point has been found to be a logarithmic function of the temperature, and if the viscosity is known at two different temperatures within this range, the viscosities at other temperatures above the wax separation point may be readily determined by platting a straight line chart through the two known viscosities on semi-logarithmic cross-section paper. The position and slope of this line will be different for different oils, and this easy means of determining the viscosity at different temperatures is of great practical benefit in connection with viscous liquid flow computations.

The comparison shown in Fig. 2 of the kinematic viscosities of water and of five typical American lubricating oils at temperatures of 100 deg. fahr. and 212 deg. fahr. is of interest both in showing the relative viscosities of these different liquids and in showing the varying effect of temperature changes.

FIG. 4.—KINEMATIC VISCOSITIES FOR ENGLER AND SAYBOLT VISCOSIMETER READINGS

Time, Engler	Time, Saybolt	Time, ratio	Engler degrees = time, Engler / 51.3[a]	Kinematic viscosity	Time, Engler	Time, Saybolt	Time, ratio	Engler degrees = time, Engler / 51.3[a]	Kinematic viscosity
Sec.	Sec.			Poises g/cm³	Sec.	Sec.			Poises g/cm³
56	32.4	1.73	1.09	0.0155	130	83.6	1.56	2.54	0.1624
58	33.7	1.72	1.13	.0208	140	90.6	1.55	2.73	.1793
60	35.1	1.71	1.17	.0259	150	97.4	1.54	2.93	.1956
62	36.5	1.70	1.21	.0308	160	104.4	1.53	3.12	.2121
64	37.8	1.69	1.25	.0356	180	117.7	1.53	3.51	.2437
66	39.2	1.68	1.29	.0403	200	131.5	1.52	3.90	.2753
68	40.6	1.68	1.33	.0448	225	148.3	1.52	4.39	.3140
70	42.0	1.67	1.36	.0495	250	165.3	1.51	4.88	.3523
75	45.5	1.65	1.46	.0603	275	182.0	1.51	5.36	.3904
80	49.0	1.63	1.56	.0709	300	199.0	1.51	5.85	.4282
85	52.4	1.62	1.66	.0810	325	215.6	1.51	6.34	.4660
90	55.8	1.61	1.76	.0905	350	232.9	1.50	6.82	.5038
95	59.4	1.60	1.85	.1003	375	249.2	1.50	7.31	.5413
100	62.8	1.59	1.95	.1095	400	266.1	1.50	7.80	.5784
110	69.8	1.58	2.15	.1278	500	333.0	1.50	9.75	.7271
120	76.8	1.56	2.34	.1453	600	400.0	[b] 1.50	11.70	.8753

[a] See T. P. 100, p. 23.
[b] This value holds good for all higher viscosities.

Pressure Affects Viscosity Slightly

Putting a liquid under high pressure also has an effect on its viscosity, but in general this effect is slight for the pressures ordinarily encountered in commercial practice, and it is usually unnecessary to take into account the increase in viscosity due to increase in pressure.

A change in pressure affects the viscosity of water very little, while the viscosities of animal, vegetable, and mineral oils are materially increased when the pressure is raised to 10,000 or 15,000 pounds per square inch.

The viscosity of oil and other sluggish liquids is determined by an instrument known as a viscosimeter of which a number of standard makes are in use at the present time. The general principle used in all viscosimeters is to heat the liquid to a definitely controlled temperature in a vessel, and when the desired temperature is attained, a small orifice in the bottom of the vessel is opened and the liquid allowed to flow into a measuring flask, the time, in seconds, required to fill the flask being the measure of the viscosity.

Computing Kinematic Viscosities

Winslow H. Herschel, associate physicist, United States Bureau of Standards, standardized the Saybolt Universal Viscosimeter in 1918, established time-ratios for converting Saybolt (American), Engler (Germany), and Redwood (British) viscosimeter readings into common terms, and tabulated the kinematic viscosities corresponding to the various viscosimeter readings. The results of his work are published in United States Bureau of Standards Technologic papers Nos. 100 and 112. A tabulation of the time-ratios and kinematic viscosities

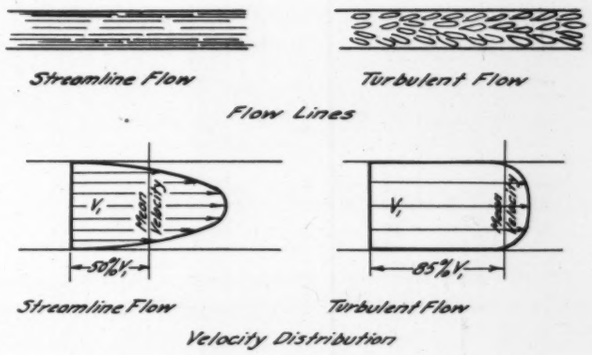


FIG. 5.—STREAM LINE AND TURBULENT FLOW

for the various viscosimeter readings as determined by the United States Bureau of Standards are shown in Figs. 3 and 4. The formulas used by the United States Bureau of Standards in computing the kinematic viscosities are as follows:

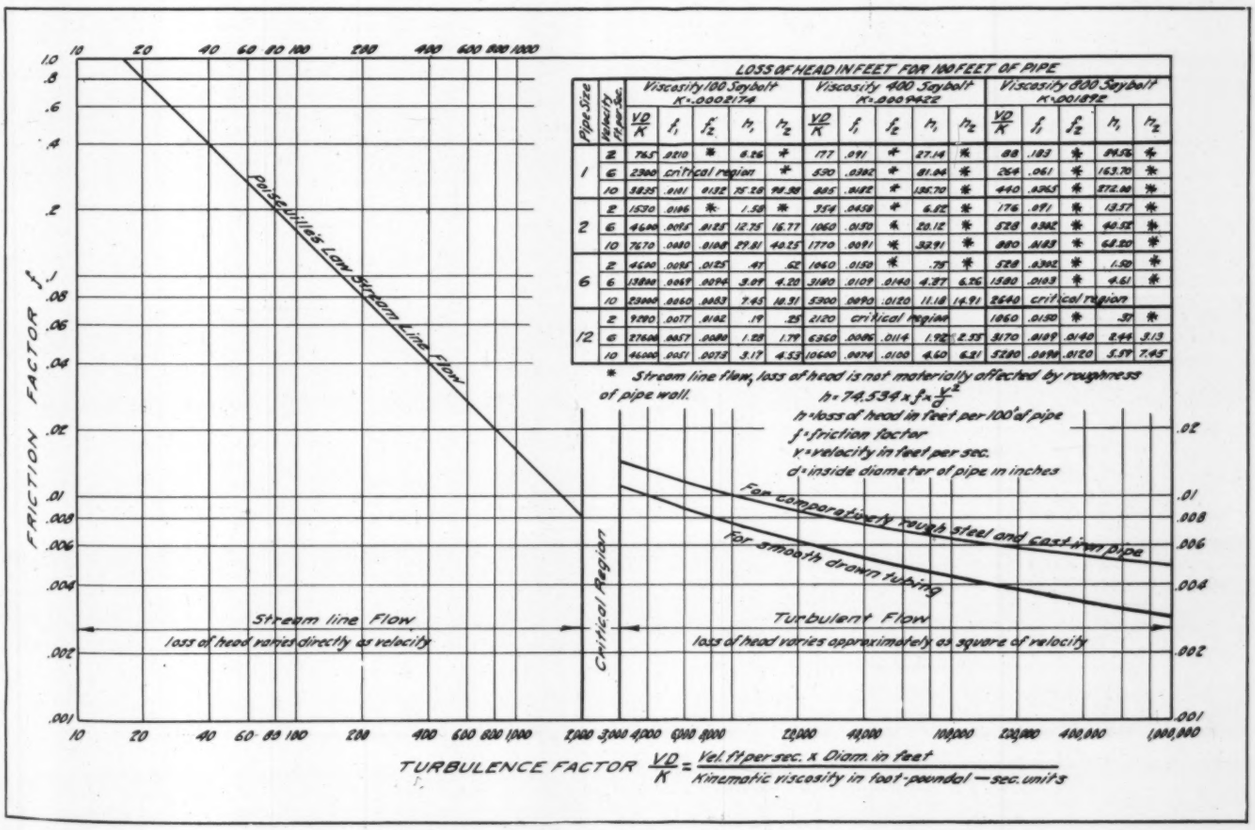
For Saybolt Viscosimeter—$k = .00220S - \frac{1.80}{S}$

For Engler " —$k = .00147E - \frac{3.74}{E}$

For Redwood " —$k = .00260R - \frac{1.715}{R}$

k = Kinematic viscosity, $\frac{\text{poises}}{\text{density in grams per cu. cm.}}$

S = Saybolt time, seconds


FRICTION FACTOR f
Poiseuille's Law System Line Flow
Critical Region
Stream line Flow
loss of head varies directly as velocity
Turbulent Flow
loss of head varies approximately as square of velocity
For comparatively rough steel and cast iron pipe
For smooth drawn tubing
TURBULENCE FACTOR $\frac{VD}{K}$ = Vel. ft per sec. x Diam. in feet / Kinematic viscosity in foot-poundal—sec. units
LOSS OF HEAD IN FEET FOR 100 FEET OF PIPE
Viscosity 100 Saybolt K=.0002174
Viscosity 400 Saybolt K=.0009422
Viscosity 900 Saybolt K=.001872
Pipe Size
Velocity Ft per Sec.
* Stream line flow, loss of head is not materially affected by roughness of pipe wall.
$h = 74.534 \times f \times \frac{V^2}{d}$
h = loss of head in feet per 100' of pipe
f = friction factor
V = velocity in feet per sec.
d = inside diameter of pipe in inches

Pipe Size	Velocity Ft per Sec.	Viscosity 100 Saybolt K=.0002174 $\frac{VD}{K}$	f_1	f_2	h_1	h_2	Viscosity 400 Saybolt K=.0009422 $\frac{VD}{K}$	f_1	f_2	h_1	h_2	Viscosity 900 Saybolt K=.001872 $\frac{VD}{K}$	f_1	f_2	h_1	h_2
1	2	765	.0210	*	6.26	*	177	.091	*	27.14	*	88	.183	*	54.58	*
	6	2300	critical region			*	530	.0392	*	81.04	*	264	.061	*	163.70	*
	10	3825	.0101	.0132	15.26	19.98	885	.0182	*	135.70	*	440	.0365	*	272.00	*
2	2	1530	.0196	*	1.58	*	354	.0458	*	6.82	*	176	.091	*	13.57	*
	6	4600	.0085	.0125	12.75	16.77	1060	.0150	*	20.12	*	528	.0302	*	40.52	*
	10	7670	.0080	.0108	29.81	40.25	1770	.0091	*	32.91	*	880	.0183	*	68.20	*
6	2	4600	.0085	.0125	.47	.62	1060	.0150	*	.75	*	528	.0302	*	1.50	*
	6	13800	.0069	.0094	3.97	4.20	3180	.0109	.0140	4.37	6.26	1580	.0103	*	4.61	*
	10	23000	.0060	.0063	7.45	10.31	5300	.0090	.0120	11.18	14.91	2640	critical region			
12	2	9200	.0077	.0102	.19	.25	2120	critical region				1060	.0150	*	.30	*
	6	27600	.0057	.0080	1.29	1.79	6360	.0085	.0114	1.92	2.55	3170	.0109	.0140	2.44	3.13
	10	46000	.0051	.0073	3.17	4.53	10600	.0074	.0100	4.60	6.21	5280	.0098	.0120	5.89	7.45

FIG. 6.—FRICTION FACTOR CHART FOR COMPUTING LOSS OF HEAD

SIZE	STD. WT PIPE		Ex. STRONG PIPE		DBL. EX. STR. PIPE	
	Wall Thickness	Inside Diam.	Wall Thickness	Inside Diam	Wall Thickness	Inside Diam
⅛	.068	.269	.095	.215		
¼	.088	.364	.119	.302		
⅜	.091	.493	.126	.423		
½	.109	.622	.147	.546	.294	.252
¾	.113	.824	.154	.742	.308	.434
1	.133	1.049	.179	.957	.358	.599
1¼	.140	1.380	.191	1.278	.382	.896
1½	.145	1.610	.200	1.500	.400	1.100
2	.154	2.067	.218	1.939	.436	1.503
2½	.203	2.469	.276	2.323	.552	1.771
3	.216	3.068	.300	2.900	.600	2.300
4	.237	4.026	.337	3.826	.674	3.152
5	.258	5.047	.375	4.813	.750	4.063
6	.280	6.065	.432	5.761	.864	4.897
8	.322	7.981	.500	7.625	.875	6.875
10	.365	10.020	.500	9.750		
12	.375	12.000	.500	11.750		
O.D. PIPE						
14	.375	13.250				
16	.375	15.250	Other wall thicknesses are			
18	.409	17.182	also used giving different			
20	.409	19.182	inside diameters			
24	.500	23.000				

FIG. 7.—INSIDE DIAMETERS OF PIPE

$$E = \text{Engler Degrees}, \frac{\text{Engler time, seconds}}{51.3}$$

R = Redwood time, seconds

Those who have occasion only once in a great while to make fluid flow computations may well be confused by the great variety of formulas in use, and particularly, by the variations in the systems of units employed. Viscosity is especially confusing because its absolute value is usually expressed in poises, centimeter-gram-second units, but it is sometimes expressed in centipoises and sometimes in foot-poundal-second units. Those who use the centipoise as a unit do so because the viscosity of water at 68 deg. fahr. is one centipoise, and this fact simplifies the computations. The kinematic viscosities computed by the United States Bureau of Standards formulas and shown in Figs. 3 and 4 are expressed in poises divided by density in grams per cubic centimeter. These values may be transposed to centipoise units by multiplying by 100 and to the English foot-poundal-second units by multiplying by .001076.

Nature of Resistance	Equivalent length of straight Pipe expressed in Pipe Diameters Safe average values
Square-Edged entry upstream end of pipe flush with inside face of reservoir wall	20*
Entry like Bordas mouthpiece upstream end of pipe projecting into reservoir.	40*
Well rounded entry	Zero*
90° Curves smooth same inside diameter as pipe.	
Center-line radius = diam. of Pipe	20*
Center-line radius = 2 to 8 diameters	10*
90° Elbows common screw end (experiments of ⅜" to 6" Elbows)	30*
Tees, common screw end full size branch (experiments of 1" to 4" Tees)	60*
Square Elbow (intersection of two cylinders)	50*
Gate Valve, wide open	5
Globe Valve, wide open	100

*Marks Handbook, Second Edition, page 278

FIG. 8.—HEAD LOSS CAUSED BY VALVES AND FITTINGS

The gravity of the oil is usually expressed in degrees Baumé as determined by the hydrometer test and this may also lead to some confusion because there are two Baumé gravity scales in use for liquids lighter than water. The relation of the original Baumé scale to specific gravity is expressed by the formula:

$$\text{Specific gravity} = \frac{140}{130 + \text{Baumé degrees}}$$

It was found, however, that in the oil industry the great majority of the hydrometers in use were not graduated on the basis of the above relation but were more nearly in agreement with a scale represented by the formula:

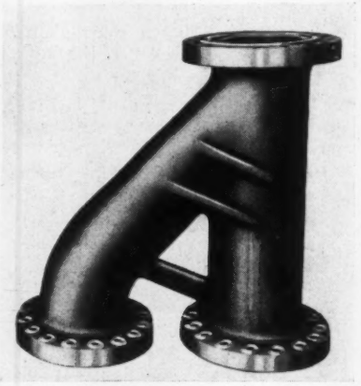


FIG. 9.—SPECIAL FITTING FOR DUPLICATING OIL PIPE LINE AT RIVER CROSSINGS

$$\text{Specific gravity} = \frac{141.5}{131.5 + \text{Baumé degrees}}$$

This scale is now known as the API scale. It was approved in December, 1921, by the American Petroleum Institute, the United States Bureau of Mines, and the United States Bureau of Standards, and it was recommended that in future no other scale should be used in the petroleum industry.

For liquids heavier than water a single Baumé scale is in general use and its relation to specific gravity is expressed by the formula:

$$\text{Specific gravity} = \frac{145}{145 - \text{Baumé degrees}}$$

Fluid Flow in Pipes

Two examples of empirical formulas for the flow of oil in pipes are Forrest M. Towl's formula, extensively used by the eastern pipe line companies, and A. F. L. Bell's formula, which is based on experiments with heavy asphaltum base California oil. These two formulas represent the two extremes of oil pipe line service in this country, but their application is strictly limited to certain specific conditions. Neither of them takes the viscosity of the liquid into account, but the Bell formula does recognize the effect of temperature on the flow characteristics. For general use in connection with viscous liquid flow computations, it is evident that a universally applicable method of computation is necessary, which takes into account the various factors affecting the flow.

FIG. 10.—CROSS-COUNTRY PIPE LINES SOMETIMES ENCOUNTER DIFFICULT CONDITIONS

The experiments of Professor Osborne Reynolds in England a half century ago demonstrated that fluids flowing in straight pipes at relatively low velocities travel with a steady stream line motion, each particle moving in a straight line, but when the velocity is increased to a certain point, known as the critical velocity, eddies begin to form, and as the velocity is still further increased the stream line flow changes to turbulent flow.

Stream Line and Turbulent Flow

Fig. 5 illustrates the two varieties of flow known respectively as stream line, viscous, or non-sinuous flow, and turbulent, eddy, or sinuous flow in straight pipes. The first two sketches show the flow lines of the individual particles making up the aggregate of the liquid, the third and fourth being typical velocity curves showing the distribution of velocity throughout the cross-section of the pipe.

In stream line flow, it will be noted that the velocity is very small near the pipe wall, increasing gradually toward the center, the ratio of the average to the maximum velocity being approximately .5; while in turbulent flow, the eddies tend to keep the velocities adjacent to the pipe wall and at the center of the pipe more nearly uniform, the ratio of the average to the maximum velocity being approximately .85.

Due to the relatively slow movement of the liquid at the pipe wall when the flow is stream line, the roughness of the bore of the pipe has comparatively little effect on the friction loss, while in turbulent flow where the velocity at the pipe wall is considerably increased, the effect of a rough interior surface is more pronounced. In commercial practice the flow of a mobile liquid like water, except in the case of very small tubes and moderate velocities, is always well above the critical velocity, resulting in turbulent flow, but in handling viscous liquids the flow is more likely to be stream line.

Professor Reynolds discovered the conditions which produce these two kinds of flow and established the

FIG. 11.—SIX LINES OF PIPE STRETCHING ACROSS THE PLAINS

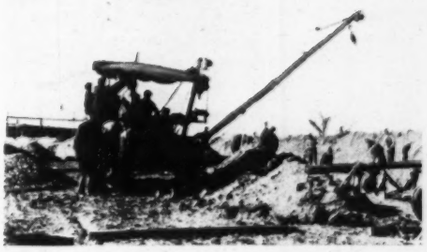


FIG. 12.—LAYING PIPE THROUGH COMPARATIVELY LEVEL COUNTRY

FIG. 13.—MODERN EQUIPMENT USED FOR INSTALLING PIPE LINE

Reynolds' criterion $\frac{VD}{K}$, in which V equals the velocity of the fluid, D equals the diameter of the pipe, and K

FIG. 14.—PIPING ASSEMBLED READY TO LOWER INTO TRENCH

equals the kinematic viscosity of the fluid. His experiments indicated that for values of $\frac{VD}{K}$ less than 2,000 the flow is stream line, and for values above 3,000 the flow is turbulent. In the critical region between 2,000

FIG. 15.—ROUTE OF PIPE LINE FOLLOWING THE TOPOGRAPHY OF THE LAND

and 3,000, the flow is uncertain and may be either stream line or turbulent. When designing a pipe line it is, consequently, desirable to change the pipe size bringing the flow into the viscous region if the computations indicate that the flow is within the critical range, or if this is impracticable, to assume turbulent flow conditions throughout the critical region, in order to be on the safe side.

A great many experiments have been conducted by numerous investigators on the flow of fluids in pipe lines and formulas galore have been proposed, many of which involve the use of arbitrary constants and decimal exponents, the values of which vary widely according to the different investigators. While such formulas accurately express the particular ranges of flow conditions on which they are based, it appears quite probable that for general use the uncertainty in selecting the proper values for the constants and exponents involves errors fully as great as would be incurred by using the simpler Chézy formula, in which the loss of head is inversely proportional to the pipe diameter, and directly proportional to the velocity squared, multiplied by a variable surface friction factor.

In 1914 Stanton and Pannell conducted a series of tests on the flow of air, water, and oil in smooth-drawn brass pipes ranging in diameter from .144-in. to 4.97-in.

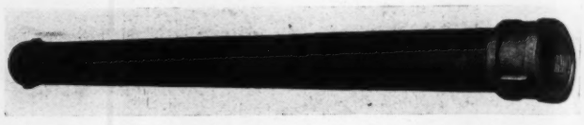


FIG. 16.—NEW FORM OF ALLOY CAST IRON PIPE WITH SCREWED CONNECTIONS

at the National Physical Laboratory in England, and from the data secured prepared a chart based on the Chézy formula for computing the flow of fluids in pipes. Their chart takes into account the various factors affecting the flow, and hence is universally applicable to fluids of all kinds both liquid and gaseous.*

The research laboratory of applied chemistry of the Massachusetts Institute of Technology in 1921 published the results of an extensive series of tests made by Wilson, McAdams & Seltzer with four widely different

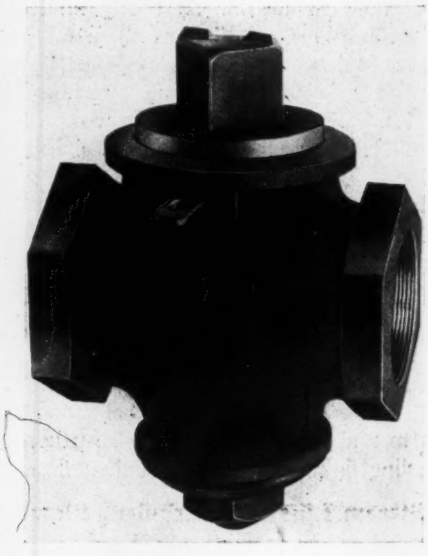


FIG. 17.—OIL COUNTRY COCK LOCK PATTERN

mineral oils ranging from 20,700 to 330 Saybolt seconds viscosity at 70 deg. fahr. and flowing through approximately 150 feet of 1-in., 2-in., and 4-in. commercial steel pipe lines. While the results of these tests check very well with the Stanton and Pannell experiments in the stream line flow region, consistently higher values for the variable friction factor were found in the turbulent

* See Philosophical Transactions Royal Society, Vol. 214, 1914; also American Petroleum Refining by H. S. Bell, C. E.

flow region. This divergence is readily accounted for by the frictional resistance caused by the increased roughness of the inside wall of the pipe line.

The results of other investigators experimenting with different fluids such as water, steam, and air flowing in pipes ranging in size up to several feet in diameter also agree very well with these data. The chart shown in Fig. 6 is based on a study of extensive data secured by various investigators from experiments with oil, water, steam and air, and consequently may be used for viscous liquid flow as well as for water, air, and gas. Allowance must be made, however, and extreme accuracy can not be expected when the liquid is in a semi-solid plastic condition or carries an appreciable percentage of solid material in suspension. Errors of considerable magnitude are also introduced when the pipe is badly corroded or the cross-sectional area reduced and the friction factor increased by incrustations or deposits of solid material. Variation in the temperature of the liquid at the center

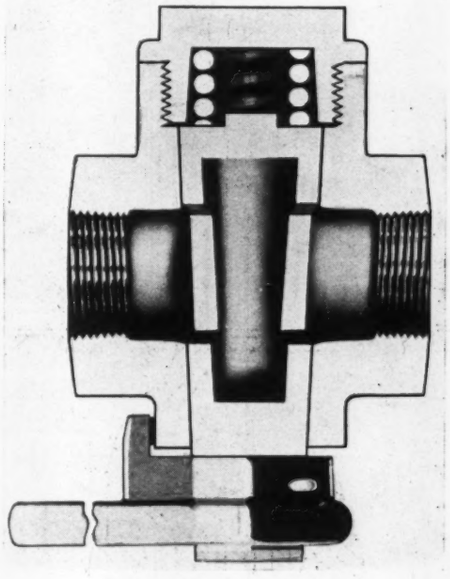


FIG. 18.—SPRING PLUG COCK

of the stream and at the wall of the pipe may also cause a pressure drop differing from the calculated results based on the average temperature, particularly when the flow is stream line and when the viscosity of the liquid increases rapidly with a relatively small change in temperature.

Friction Factor Chart

In Fig. 6 it will be noted that throughout the stream line flow region the friction factor curve is a single straight line, platted at an angle of 45 deg. with the axes of coordinates, while in the turbulent region two curves are shown, the lower curve approximating the values proposed by Stanton and Pannell based on experiments with smooth drawn brass pipes, and the upper curve being slightly higher than the one proposed by the Massachusetts Institute of Technology for commercial steel and cast iron pipe lines. In this connection it is of interest to note that for the larger turbulence factor values, the upper curve is also very close to the Williams and

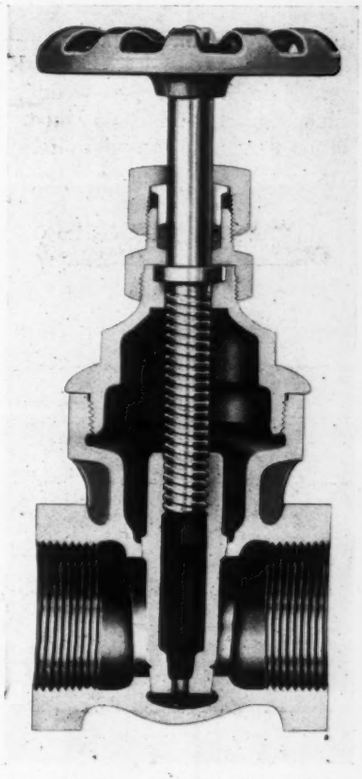


FIG. 19.—BRASS WEDGE GATE VALVE

Hazen Tables, with "C" equal to 120, which is the value given for cast iron pipe lines five years old.

The X-axis, or horizontal scale, of the chart is the

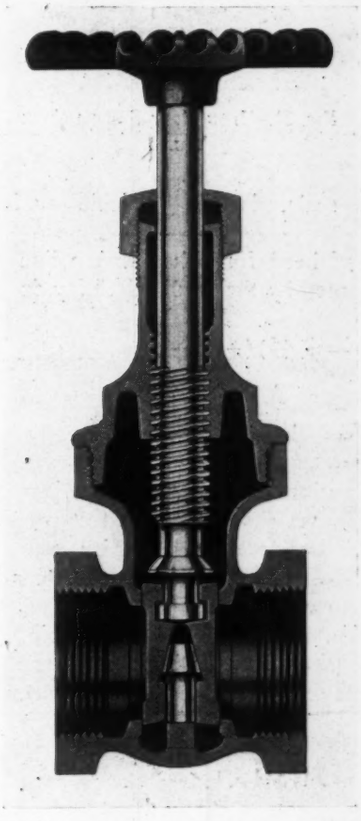


FIG. 20.—BRASS DOUBLE DISC GATE VALVE

Reynolds' criterion or turbulence factor, $\frac{VD}{K}$, in which V = velocity of fluid in feet per second, D = inside diameter of pipe in feet, and K = kinematic viscosity of the fluid in foot-poundal-second units. The Y-axis,

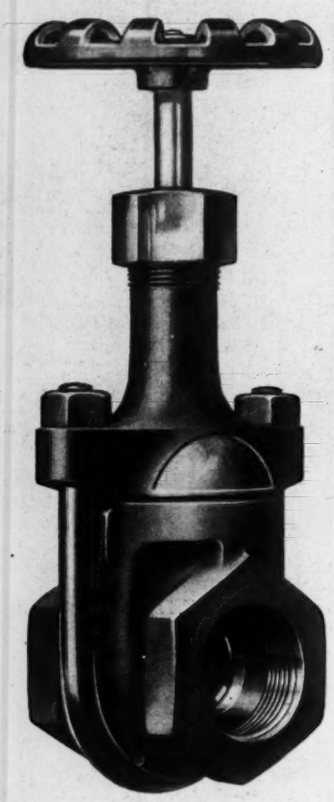


FIG. 21.—IRON BODY SADDLE GATE VALVE

or vertical scale, is the variable surface friction factor "f".

The loss of head for either stream line or turbulent flow is computed by the modified Chézy formula:

$$H = 4 \times f \times \frac{L}{D} \times \frac{V^2}{2g}$$

in which H = loss of head in feet
f = friction factor
L = length of pipe in feet
D = inside diameter of pipe in feet
V = velocity of fluid in feet per second
g = acceleration of gravity
(32.2 feet per second, per second)

In order to express the formula in more convenient units, let us assume a unit length of pipe of 100 feet, express the pipe diameter in inches, and collect all of the constants into one factor. The formula then becomes:

$$h = 74.534 \times f \times \frac{V^2}{d}$$

in which h = loss of head in feet per 100 feet of pipe
d = diameter of pipe in inches

The loss of head values for various conditions shown in tabulation at upper right hand corner of chart were computed by the aid of the chart and indicate the effect of high viscosity on the friction loss in pipes, particularly in the smaller pipe sizes, and in the stream line flow region where the friction loss varies directly with the viscosity. In the turbulent region the effect of viscosity is relatively less, possibly due to increased violence of the eddy currents as the liquid becomes more mobile. For this reason, when the flow is turbulent considerably less friction loss is saved by heating the liquid to reduce its viscosity than when the flow is stream line. This fact may be illustrated by the data shown in table Fig. 6. It will be noted that the flow is stream line at both 400 and 800 Saybolt seconds viscosity for a 10 foot velocity in a 2-in. pipe, and turbulent for a 10 foot velocity in a 12-in. pipe. The values shown in the table indicate that

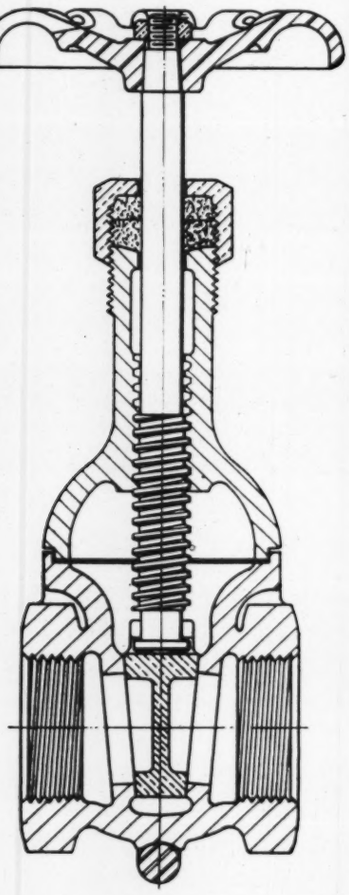


FIG. 22.—IRON BODY SADDLE GATE VALVE—SECTIONAL VIEW

the saving in loss of head by lowering the viscosity from 800 to 400 is over 50 per cent in the first case and in the second case is less than 18 per cent.

At first thought it would appear that the same formula could not be used for both stream line and turbulent flow, because in stream line flow the head loss is known to vary directly as the velocity, while in turbulent flow it varies approximately as the square of the velocity. It was discovered, however, by Stokes in England, about 70 years ago, and later rediscovered by several other in-

vestigators, that in stream line flow the friction factor "f" in Chézy's formula varies inversely as the velocity and the V^2 consequently reduces to V as in Poiseuille's Law. In turbulent flow the friction factor is quite evidently a more complicated function and does not plat as a straight line on logarithmic paper. This curve is also raised or lowered by variation in the roughness of the interior surface of the pipe and a certain amount of judgment has to be used in selecting proper values for "f" in this region. The two curves shown in the chart may be taken as representing the upper and lower limits encountered in ordinary commercial practice and the value chosen for "f" should lie somewhere between the two curves, depending upon the relative roughness of the interior surface of the pipe.

Applying the Chart

In order to illustrate the method of applying the chart to an actual problem, let us assume that it is required to transport 30,000 barrels of mineral oil per 24 hours, the oil having a Baumé gravity of 27 deg. and a viscosity of 130 Saybolt seconds at the average atmospheric temperature of 85 deg. fahr. Let us also assume that it is planned to place the pumping stations 40 miles apart and to carry 600 pounds per square inch pressure at the pump discharge with provisions for later increasing the pressure to secure additional pipe line capacity. It is desired to find the commercial pipe size which will satisfy these conditions.

First, convert the known data into the units used on the chart:

Density

$$\text{Specific gravity} = \frac{141.5}{131.5 \times 27 \text{ (Baumé degrees)}} = .8927$$

Density $= .8927 \times 62.43$ (weight of water) $= 55.73$ lbs. per cu. ft.

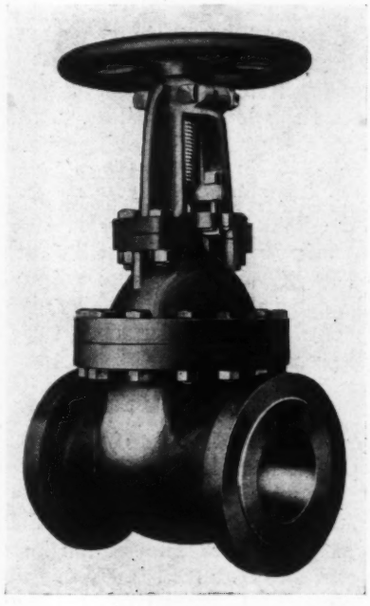


Fig. 23.—Extra Heavy, Iron Body, O. S. & Y., Flanged Wedge Gate Valve

Pressure Drop

$$\frac{600 \text{ lbs. per sq. in.}}{\frac{40 \text{ miles} \times 5280 \text{ feet}}{100 \text{ feet}}} = .2841 \text{ lbs. per sq. in. per 100 ft. of pipe}$$

$$h = \frac{.2841 \times 144 \text{ sq. in.}}{55.73 \text{ (density)}} = .7341 \text{ head loss in feet per 100 ft. of pipe}$$

Quantity of Oil Flowing

$$\frac{30000 \text{ bbls. of oil}}{24 \text{ hrs.} \times 60 \text{ min.} \times 60 \text{ sec.}} = .3472 \text{ bbls. per sec.}$$

$$\frac{.3472 \times 42 \text{ gallons per bbl.}}{7.48 \text{ gallons per cu. ft.}} = 1.950 \text{ cu. ft. per sec.}$$

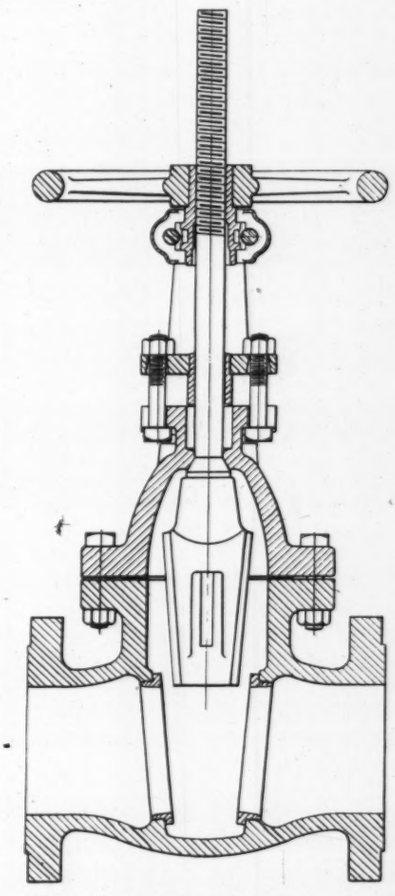


Fig. 24.—Series 600, Cast Steel Ball Bearing Yoke, Flanged Wedge Gate Valve (Sectional View)

Kinematic Viscosity

The kinematic viscosity may be interpolated from the data shown in Fig. 4 or may be computed from the United States Bureau of Standards formula.

$$k = .0022 \times 130 - \frac{1.80}{130} = .2722 \text{ in C. G. S. units}$$

To convert to English units multiply by .001076

$$k = .2722 \times .001076 = .002929$$

When the pipe size and other conditions except pressure drop are known, the loss of head per 100 feet of

pipe may be computed directly from the modified Chézy formula, but when the pipe size is unknown the situation becomes more complicated, and the usual method is to proceed by trial and error. First, assume a pipe size and compute the pressure drop, then by comparing the computed value with the desired result estimate a closer pipe size and recompute, continuing in this manner until the specified conditions are fulfilled. Let us then assume a 12-in. pipe and compute the pressure drop:

$$V = \frac{1.95 \text{ cu. ft. per sec.}}{.7854 \text{ sq. ft. (cross-sectional area of pipe)}}$$

$= 2.48$ ft. per sec.

From tabulation in Fig. 6, we see that for 12-in. pipe, 2 foot velocity, and 100 Saybolt viscosity, the head loss would be somewhere between .19 and .25 feet per 100 feet of pipe. Comparing this with the anticipated head loss of .73, it is apparent that the assumed 12-in. pipe is too large.

Next taking 10-in. pipe, the next smaller commercial size, we have:

$$V = \frac{1.95}{.5454 \text{ sq. ft.}} = 3.57 \text{ ft. per sec.}$$

$$\frac{VD}{K} = \frac{3.57 \times 10/12}{.0002929} = 10400$$

$f_1 = .0075$ and $f_2 = .0099$

$$h_1 = 74.534 \times .0075 \times \frac{3.57^2}{10} = .71 \text{ ft., head loss per 100 feet of pipe}$$

$$h_2 = 74.534 \times .0099 \times \frac{3.57^2}{10} = .94 \text{ ft., head loss per 100 feet of pipe}$$

The specified head loss of .73 feet falls between the maximum and minimum computed values and 10-in. pipe should be used.

The diameter used in flow computations is, of course, the inside diameter of the pipe and for standard weight steel pipe in sizes 1 to 12-in., inclusive, the nominal pipe size is very close to the actual inside diameter, and may be used in flow computations without appreciable error. Originally, the bore of standard weight steel pipe in sizes smaller than 1-in. also conformed to the nominal pipe size, but the walls of these pipes were thicker than necessary and when the wall thicknesses were later reduced this was accomplished by increasing the inside diameter, because the outside diameters were fixed by the established threaded and flanged joint connections. Changing the outside diameters would have resulted in expensive replacement of tool equipment at the manufacturing plants and in great confusion in industry, due to lack of interchangeability, so the inside diameters were changed instead, and in the present standard weight steel pipes smaller than 1-in. the bore is larger than the nominal pipe size. Steel pipe in sizes larger than 12-in. is designated by its outside diameter and is known as O. D. pipe. Extra heavy and double extra heavy pipes have considerably thicker walls and smaller inside diameters than standard weight pipe, so it is evident that nominal pipe sizes can only be used to a limited extent in flow computations. The table shown in Fig. 7 shows inside diameters of standard weight, extra heavy, and double extra heavy steel pipe, sizes ⅛ to 12 inches inclusive, and of O. D. pipe, sizes 14 to 24 inches inclusive. Sizes 3½, 4½, 7, 9, 11, 15 and 22 inches are omitted, in accordance with the simplification program now being carried out in the pipe, valve and fittings industries whereby sizes, the demand for which does not warrant production, are being eliminated.

Pressure Drop Through Valves and Fittings

The excess loss of head caused by valves, fittings changing the direction of flow or changing the cross-sectional area of the pipe, branch outlets splitting the flow of a single pipe into two or more separate streams, and the like, must also be taken into consideration in computing the pressure drop in a pipe system. The standard engineering handbooks give values for head losses of this character expressed in terms of the number of pipe diameters required to produce an equivalent loss, and these values are applicable to viscous liquids as well as water, because the unit used is affected proportionately with the valve or fittings by changes in the viscosity of the liquid thus eliminating viscosity from consideration. The table shown in Fig. 8 may be used for this purpose, the equivalent length of pipe represented by the various valves and fittings incorporated in the pipe line being added to the actual length of the pipe in order to determine the total pressure drop.

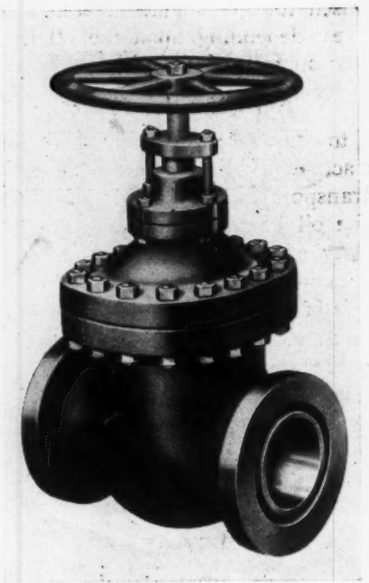


Fig. 25. 1,600 Lb. Test Iron Body, N. R. S., Screwed End Double Disc Gate Valve

Installation and Maintenance

In this brief article it is only possible to discuss the problems of proper installation and maintenance of piping for viscous liquids in a very general way, but it is hoped that the principles outlined will be helpful in studying the requirements of a specific installation. In general, the problems connected with the piping of viscous liquids are similar to those for other pipe lines carrying liquids, except that certain features, such as protection against chilling or freezing and provision for cleaning out deposits of solid material are greatly accen-

tuated. Pipe lines should be laid with ample slope for drainage, when this is required; ample supports and anchors should be provided; proper provisions should be made for expansion and contraction; and the pipe lines should be laid out as straight and direct as possible, eliminating unnecessary turns and other features causing avoidable pressure drop.

Provision for Clean-Out

Some viscous liquids are particularly troublesome to handle because they tend to precipitate solid material at certain temperatures which either clings to the inner walls of the pipe or collects at low points, obstructing flow and reducing the capacity of the pipe line. When conditions of this kind are encountered the design should

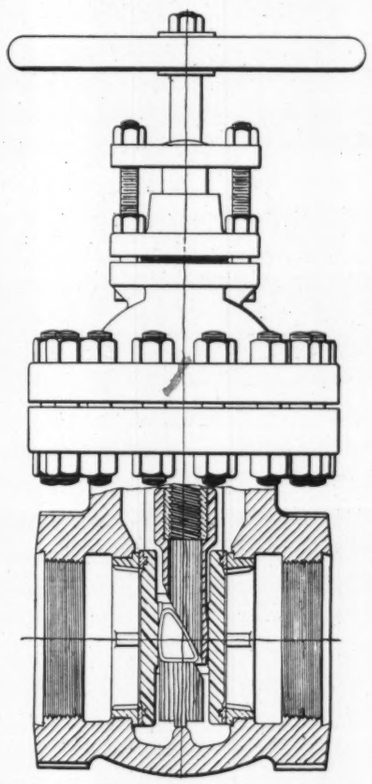


FIG. 26. 1,600 LB. TEST IRON BODY, N. R. S., FLANGED DOUBLE DISC GATE VALVE—SECTIONAL VIEW

provide for cleaning out the line when necessary by flushing, or steaming, or by the use of scrapers or other suitable means in order to maintain the efficiency of the piping system. Clean out openings should be provided at turns, pockets, etc., to facilitate the cleaning process and the pipe line should be so constructed that it can be readily dismantled when necessary for purposes of cleaning or replacement of parts.

Important sections of the piping system that are not readily accessible for repairs, are sometimes installed in duplicate with proper valve controls so that one of the lines may be held in reserve for emergency use, in case anything should happen to the other line.

The 10-in., Series 600 cast steel flanged lateral shown in Fig. 9 is a special fitting used for duplicating a cross-country pipe line at river crossings, where repairs to the line would be difficult and would require considerable time. This fitting conforms to the requirements of the American Standard for Steel Pipe Flanges and Flanged Fittings B16e, 1927, and is rated for 600 lb. per sq. in. working steam pressure at 750 deg. fahr. and for 1000 lb. working pressure when used on cold oil, water or gas.

Heating the Liquid

Heating the liquid to lower its viscosity and reduce the power required for pumping is often an economic necessity, and in some cases liquids which solidify completely at the temperature of the surrounding atmosphere are successfully handled in pipes by consistently maintaining an elevated temperature in the pipe line. The heating is usually accomplished by means of some form of heat exchanger, and the heat may be supplied either in the form of primary heat from a fuel fired heater or as secondary heat from electricity, steam, hot gases or liquids, depending upon the requirements of the problem and the availability of sources of heat supply. When a large amount of heat is necessary sound engineering requires that the heat shall be secured at low cost, and waste heat should be utilized whenever possible. Having heated the liquid to reduce its frictional resistance to flow, it is of course equally important to maintain its high temperature as long as possible and the pipe line should consequently be well insulated. In fact, it is sometimes found necessary or at least desirable, to add heat throughout the entire length of the pipe line which may be done by running a small steam pipe along side the viscous liquid line, encasing both pipes within a suitable insulating covering. A more effective exchange of heat may be accomplished by placing the steam pipe inside the viscous liquid pipe, but this construction interferes with the flow of the viscous liquid and greatly complicates the means of taking care of expansion and contraction. Specially designed electric heating systems extending along the entire length of the pipe line are also employed in some instances where accurate control of the temperature is essential.

A fortunate circumstance which frequently occurs in industry is that viscous liquids are often heated to a high temperature during manufacturing processes, and in such cases sufficient heat may often be retained by proper insulation of the pipe line to keep the viscosity within satisfactory bounds, so that further addition of heat is not required. Cross country oil pipe lines located in tropical or semi-tropical regions absorb heat from the sun and the surrounding atmosphere, and consequently are usually laid on the surface of the ground, but in regions where the atmospheric temperature may be low enough at times to materially increase the viscosity of the oil it is customary to lay the pipe in a trench and cover it with about 18-in. of soil. The oil industry, in fact, presents one of the major viscous liquid piping problems in this country, and the views shown in Figs. 10, 11, 12, 13, 14 and 15 give a general idea of the conditions encountered and the methods employed in constructing cross-country pipe lines.

Corrosion

Corrosion, the ever present enemy of metals, is one of the most important causes of waste with which the world has to contend, and in recent years an increasing amount of laboratory and field research is being devoted to the development of methods for progressively

reducing this outstanding evil. The problem of combating corrosion on the inside of the pipe is perhaps the most common one in commercial installations, but external corrosion of pipes exposed to corrosive atmospheric conditions or buried in chemically active soils is also encountered very extensively. Viscous liquid pipe

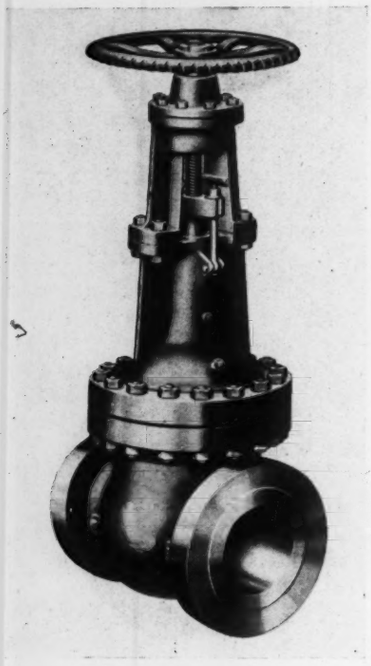


FIG. 27. SERIES 600, CAST STEEL, BALL BEARING YOKE, FLANGED WEDGE GATE VALVE

lines are not immune from these conditions and when they are present provision should be made to meet the requirements of the situation. The usual methods employed to reduce the bad effects of corrosion are:

1. Providing excess metal thickness to compensate for loss by corrosion.
2. Using metallic, bituminous, or other forms of corrosion-resistant coatings to protect the base metal.
3. Using special metals or alloys that are highly resistant to corrosion under the particular service conditions in question.
4. Chemical treatment of the fluid carried, to reduce its corrosive activity.

Any expedient of this character necessarily involves an increase in the first cost of the piping valves and fittings or additional cost for special equipment to treat the fluid, but such additional expense may be readily justified if the overall cost extending over a period of years is substantially reduced.

Numerous kinds of pipe are in commercial use, but for the average conditions found in service the use of ungalvanized steel pipe is common practice. Cast iron pipe of the bell and spigot type is also very extensively used because of its resistance to alkaline and certain forms of acid corrosion as well as its ability to withstand atmospheric and certain types of soil corrosion. Quite recently a new form of cast iron pipe has become available, the outside and inside diameters of which conform to extra heavy steel pipe dimensions and the joint connections are made with American standard pipe threads. This form of cast iron pipe may be cut, threaded and installed in exactly the same manner as steel pipe, so that it can be readily used in many places. The pipe is made from a chrome-nickel alloy cast iron. Fig. 16 shows a length of this new alloy cast iron pipe with an alloy cast iron coupling on one end and a thread protector on the other.

While first cost is usually a matter of major significance, the kind of piping material to use for a specific installation should not be determined on this basis alone, because efficiency in operation, satisfactory service, and long life are also vital factors in the overall cost and have a continuing effect on the economy of manufacturing processes. The decision, therefore, should rest upon conclusions drawn from a judicious balancing of all these factors and the most economical system is often found to be relatively high in first cost but materially lower in maintainance and operating cost.

Valves

The proper selection of valves for viscous liquid pipe lines is a subject worthy of careful consideration, and good judgment must be used if efficient operation of the

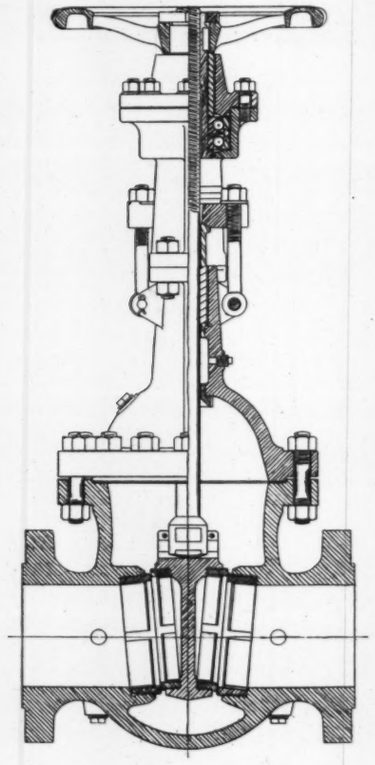


FIG. 28. SERIES 600, CAST STEEL, BALL BEARING YOKE, FLANGED WEDGE GATE VALVE

system is to be secured. The turbulent flow conditions set up by a globe valve causes head loss, and the horizontal diaphragm forms a pocket in which solids may collect and which can not be readily cleaned. The gate valve, on the other hand, affords a straight way passage

for the flow and for cleaning the line when the gate is lifted clear of the valve port. Cocks also provide a straight way passage for flow and are suitable for use with viscous liquids. Fig. 17 shows a lock pattern oil country cock which may be locked either in the open or the closed position. Fig. 18 shows the construction known as a spring plug cock in which the plug is pressed against the body seat by a spring. This construction eliminates the possibility of the plug becoming loose in service and of subsequently being tightened too hard by a careless workman. If the plug should stick it may be loosened by a sharp blow on the top and it will be immediately forced back against its seat by the pressure of the spring. Numerous patterns of cocks for various pressures, made in all iron, iron body with brass plug, and all brass constructions are available on the market.

Double Disc Gate Valve

Gate valves are of two general types, namely, solid wedge and double disc. The solid wedge gate valves are perhaps used most extensively but the double disc type valves are also very popular. The outstanding advantage of the solid wedge type is its simplicity in design. It is compact and rugged, has the least possible number of wearing parts, and is well adapted to withstanding incrustations, corrosion and hard usage. The principal advantages of the double disc valve are flexibility and ease of repairing seats. The two discs in this valve are free to adjust themselves to fit the body seats, whereas in the solid wedge valve the tapered wedge must fit the tapered body seats very accurately or the valve will leak. It is a comparatively simple machining operation to repair the seats of a double disc valve and this may be done quite satisfactorily on an ordinary lathe, but it is a difficult matter to repair the seats of a wedge type valve outside of a properly equipped valve factory. On the other hand, the double disc valve has more wearing parts and the proper functioning of the mechanism is more likely to be interfered with by corrosion and deposits of solids. Skillful design, suitable materials and high grade workmanship will produce a very satisfactory valve of either type, and in general these essential features are of greater importance than the type of the valve.

Both types of gate valves are made in two patterns, one of which has the stem thread inside the valve body and is usually called a non-rising stem (N. R. S.) valve, while the other has the stem thread outside and is usually called an outside screw and yoke (O. S. & Y.) valve.

Screwed end valves are made in sizes from ⅛-in. to 12-in. and are extensively used in sizes from ⅛-in. to 3-in. Flanged valves are usually specified in the larger sizes, but are seldom used in sizes below 2-in.

Fig. 19 shows a typical brass gate valve of the wedge type and Fig. 20 is a double disc brass gate. Both types are used extensively in sizes up to 3-in. Figs. 21 and 22 show exterior and interior views of a saddle gate valve which may be secured in the all iron construction or iron body brass mounted. Figs. 23 and 24 show exterior and interior views of an extra heavy iron body wedge type valve rated for 250-lb. working steam pressure and is typical of this class of valve. Valves of this type are also available for 25-lb. and 125-lb. working steam pressures and for 800-lb. hydraulic pressure. Figs. 25 and 26 show exterior and interior views of a 1600-lb. test, iron body double disc gate valve rated for 800-lb. cold water, oil, and gas working pressure. This type is also available in standard weight and in 700-lb. and 1000-lb. test patterns. Figs. 27 and 28 show exterior and interior views of a series 600 cast steel wedge gate valve. Similar valves for 150, 300, 400, 900, and 1350-lb. working steam pressures are also available. The cold oil, water and gas ratings for these different lines of steel valves are: series 150—300-lb., series 300—500-lb., series 400—750-lb., series 600—1000-lb., series 900—1500-lb., series 1,350—2250-lb.

Space does not permit a detailed discussion of all the different types of valves and their adaptability to the great variety of service conditions that may occur in practice, but in general only steel valves should be used for high temperature service. The upper limit of working temperature for cast iron valves is usually held at 450 deg. fahr. and for high grade bronze valves at 550 deg. fahr. The physical properties of steel, with its great strength and high ductility, make it an excellent material for valve construction, and steel valves are being used to an increasing extent even for moderate temperature service, particularly where the fire hazard is great or dangerous liquids are being handled.

Conclusion

The large field which the writer has attempted to cover in this paper has made it necessary to keep the discussion more or less general, but it is hoped that the chart for computing the flow of viscous liquids may be found useful and that the importance of the proper selection of piping, valves and fittings has been sufficiently emphasized.

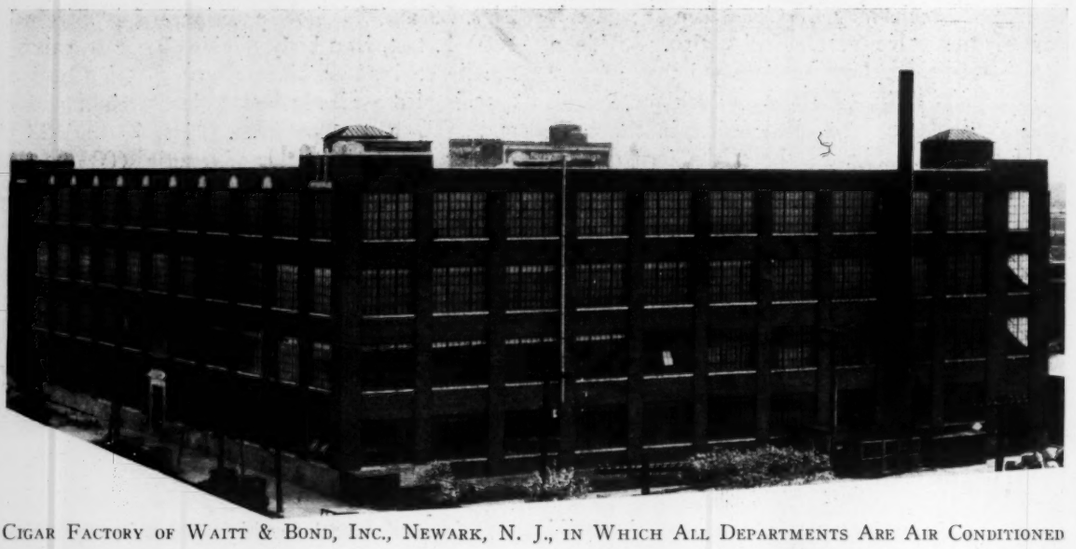
A very careful study of each individual installation is necessary in order to secure an efficient and economical piping system and there can be no doubt that the benefits to be gained in more satisfactory performance, lower operating costs, and longer life will well repay the expense and delay involved in painstaking consideration of all the various features of a piping layout.

Refrigeration Touches Many Branches of Engineering

It is common among engineers to assume that refrigeration is a distinct branch of engineering. A little thought, however, will reveal the fact that refrigeration is but one branch of the broad field of thermal engineering. It is true, of course, that nearly every branch of engineering has some contact with the growing science of refrigeration.

Among subjects lately discussed by refrigeration engineers are the use of a Diesel engine drive for compressors, new types of valves, carbon dioxide ice, the testing of unit refrigerators, new developments in refrigeration, cooling in factories for increasing production and the enormous amount of electric power used in this country in refrigeration.

Electrical, metallurgical, chemical, mechanical as well as architectural engineers have all contributed to the growth of this new branch of the science of thermodynamics.



CIGAR FACTORY OF WAITT & BOND, INC., NEWARK, N. J., IN WHICH ALL DEPARTMENTS ARE AIR CONDITIONED

Air Conditioning a Modern Cigar Factory

By Leslie P. Hale

AIR conditioning is now recognized as performing a definite economic service for industry. There are literally hundreds of industries which are affected seriously by adverse conditions of temperature, humidity, air cleanliness and air movement, and the application of this modern science of air conditioning to these industries gives the user the means by which he may remain master of the manufacturing situation at all times, insofar as atmospheric conditions are concerned.

Even more important is the ability of air conditioning to control the condition of the raw materials, enabling the manufacturer to perfect his production processes and to employ many remarkable mechanical aids. And with indoor weather thus controlled, deviations from the quality standards of the product due to rising and falling temperature and humidity are eliminated.

We here paint a sketchy picture of what this science is doing for only one important industry—the manufacture of cigars. But it must be realized that what air conditioning is doing for cigars it can do, and is doing for hundreds of other industries. We have chosen tobacco as furnishing a splendid example of what the air conditioning engineer can do for the manufacturer with a material so extremely sensitive to variations in temperature and humidity.

There is a long story between the first few handmade cigars rolled daily by the Balkan and Russian immigrants from whom most of our expert cigar makers were drawn, and the super-production of today's marvelous machines which are capable of turning out thousands of cigars every working day—cigars so far superior in every respect to those so skilfully, albeit laboriously, rolled by those old master craftsmen, that any one is the same as any other of the same brand as to quality, blend, color and taste.

The author says, "We here paint a sketchy picture of what air conditioning is doing for only one important industry—the manufacture of cigars. But it must be realized that what air conditioning is doing for cigars it can do, and is doing, for hundreds of other industries. We have chosen tobacco as furnishing a splendid example of what the air conditioning engineer can do for the manufacturer with a material so extremely sensitive to variations in temperature and humidity."

Even though our immigration laws had not restricted the entry of these foreign trained workers—even though there had not been any subsequent labor troubles in the growing tobacco industry—today's enormous consumption of cigars, which has increased so materially since the slump immediately following the war, would spell the economic death of the old-fashioned hand method. For in order to keep production abreast of this ever-increasing demand for cigars—good cigars—it was imperative that the machine supplant the hands and brains of the cigar makers.

Large Production Due to Air Conditioning

These machines, capable of turning out from eight to ten perfectly made cigars each minute, made possible this vast production—a production of well over eight billion cigars in the United States during 1928. And most of this enormous quantity is manufactured without the sacrifice of a single element of quality.

This is due, however, not alone to the cigar machines themselves, marvelous though they may be, but in a very large part to the atmospheric conditions to be found in the various departments of the factories, and, subsequently, the condition of the filler, binder and wrapper to be used in the cigars. For the machine, although it almost may be said to be possessed of human brains, is not able to decide as to the condition of the filler to be used, which of the leaves to utilize and which to discard; it must make into cigars rapidly, automatically, unthinkingly, all filler which is fed into it, regardless of condition. Such is its economic place in industry.

Tobacco Highly Hygroscopic

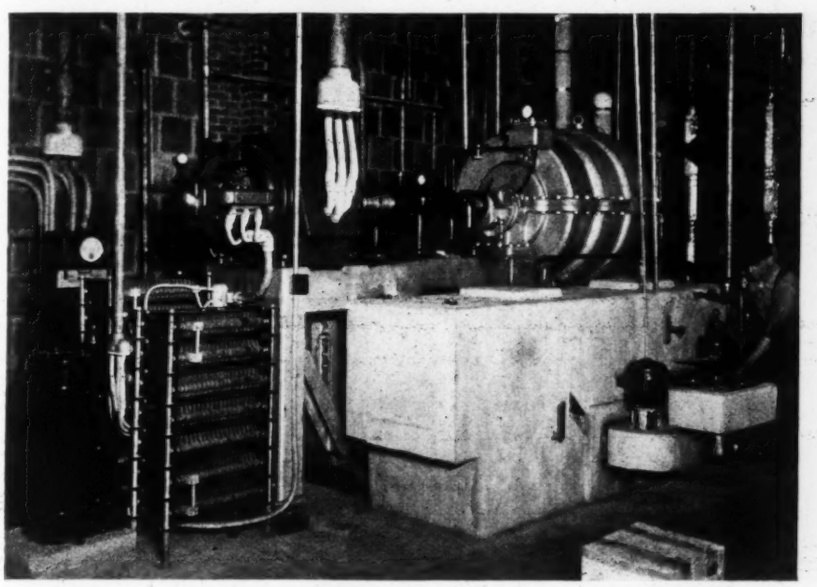
And here again it must be remembered that these machines are dealing with a vegetable product which is readily susceptible to injury and deterioration in handling and manufacturing, which is extremely sensitive to changes in temperature and humidity. For tobacco is a highly hygroscopic material and because of this, in every phase of its culture and manufacture, weather is the dominant factor.

Thus is the manufacturer whose aim is quality and quantity production almost entirely at the mercy of weather, and here, then, is an engineering problem of the highest order; to provide weather, or indoor atmospheric conditions, exactly suited to each phase of the handling and manufacture of this sensitive material, in order to assure high-speed production of a uniform and standardized product. Rather may it be said that the air conditioning engineer improves the climate in which tobacco products are manufactured, for this is exactly what he does.

Contrary to the popular and firmly-rooted belief that the climate peculiar to the tropics, particularly that of Porto Rico and Cuba, famous for good cigars, is the perfect climate for cigar manufacturing, this is not the case. There is no perfect climate, day in and day out, for manufacturing cigars or anything else. Even the tropical climate can, and must, be improved in order to speed up production sufficiently to meet the demands of today's thousands of cigar smokers.

It is quite true that tropical conditions are generally conducive to fine cigar manufacture, on account of the high relative humidity, and because there are no severe drops in temperature, the range being decidely more normal than that in New York City during the average summer. The difficulty with the tropical climate, however, is that it creates too high humidities at times, and here is where air conditioning is employed to reduce the humidity to that actually required, and, in so doing, regulates the exact humidity automatically, regardless of any sudden changes outdoors.

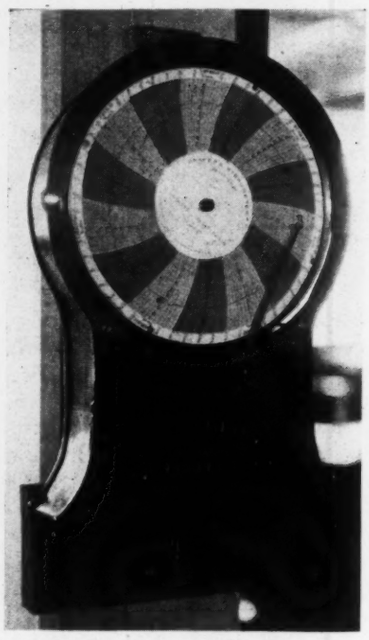
So it remained for air conditioning engineers to bring the desired atmospheric conditions into the



Centrifugal Refrigeration Machine Which Meets All the Cooling and Refrigeration Requirements. Note Its Compactness

modern cigar factory, and air conditioning, accordingly, today solves the weather problems of the world's largest cigar and cigarette manufacturers.

To the early cigar maker, weather conditions meant little or nothing. He knew that some leaves were properly conditioned and that others were not, that some contained exactly the correct amount of moisture to assure a palatable, even-burning, sweet-smelling cigar. Your early skilled cigar maker selected and rejected with the deftness of long experience, simply by feeling the leaves.



Recording Hygrometer Which Gives a Record of the Relative Humidity Maintained in Various Departments. Note Uniformity Maintained

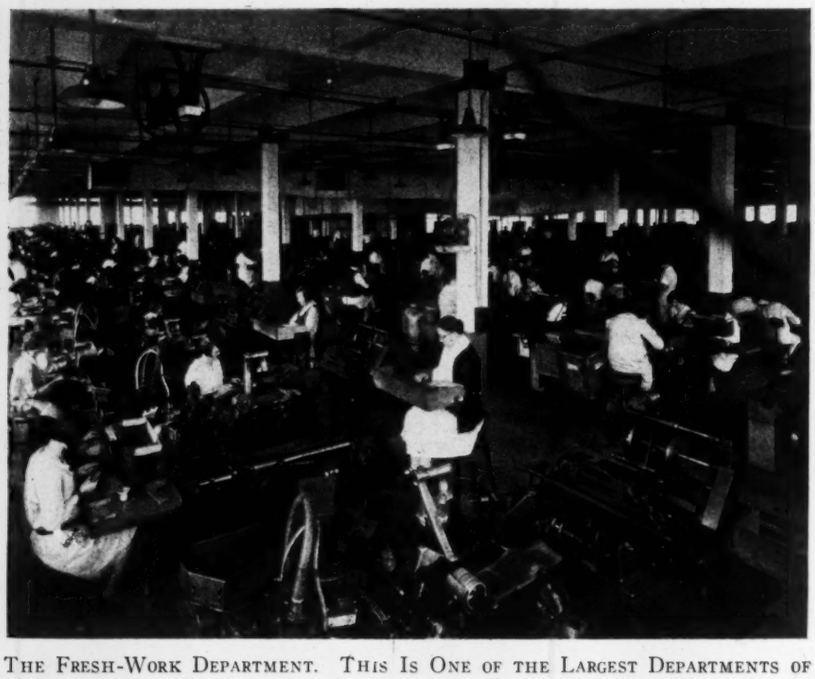
Conditioning the Filler

Since the cigar machine must use any and all filler, the problem, then, in the modern factory, is to condition the filler thoroughly and uniformly before it reaches the machine, and to hold it at this ideal condition throughout each successive operation until it leaves the factory. The filler must be neither too wet nor too dry, but must contain exactly the correct amount of moisture to make springy cigars. For even the best tobacco, improperly conditioned, cannot make a good cigar. Tobacco to be made into cigars and cigarettes must be uniformly soft and pliable, and there is an ideal moisture content at which the leaf is pliable and retains the mellow flavor and color imparted by the original curing.

Let us consider, briefly, the atmospheric needs of the modern factory turning out some five hundred to six hundred thousand cigars a day. First of all, conditioned air must be supplied to every department, every floor, at the temperature and humidity exactly suited to the work conducted in each department; each individual leaf must be brought to, and held at, the proper condition for each successive operation from blending of filler to shipment; mold spores must be prevented from forming on the cigar; superfluous moisture must be removed from all fresh made cigars, due to the soft binder, the wet wrapper and the paste on the heads, and the humidity subsequently must be held at the correct point; the cigars must be seasoned, packed and stored under absolutely constant atmospheric conditions; the storage department, in fact, is really a giant humidor in which, under the controlled atmospheric conditions, cigars may be kept almost indefinitely, as far as "condition" is concerned, without any change whatsoever.

It will thus be seen that the large production cigar manufacturer is primarily at the mercy of the weather, and thus largely dependent upon the skill of the air conditioning engineer.

An interesting and excellent example of an air conditioning system installed in the plant of a cigar manufacturer making high grade cigars on a very large scale, is that in the thoroughly



The Fresh-Work Department. This Is One of the Largest Departments of Its Kind in the World, Having a Capacity of More Than a Half-Million Cigars a Day

modern new plant of Waitt & Bond, Inc., at Newark, N. J. In this factory, which has a capacity of over a half-million cigars a day, the air conditioning installation is complete in every respect, consisting of equipment for air conditioning, including refrigeration, for drying and for processing.

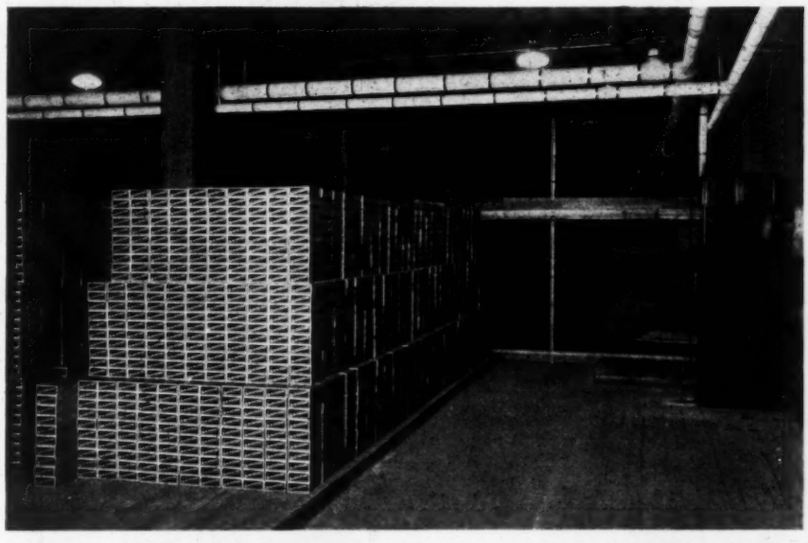
Three Sources of Supply for Conditioned Air

There are three main sources of supply from which the conditioned air is delivered to the various departments; a central station system, an ejector dryer and a unit air conditioner. A brief description of each of these three main sources will be given later.

Departments on every floor of the factory are air conditioned. On the first floor there is the storage or seasoning department, in which, as the name implies, the finished, boxed cigars are stored for seasoning prior to shipment. On the second floor is the stripping department, where the stems are carefully stripped from the delicate leaves. The entire third floor is devoted to manufacturing, the actual making of the cigars, and includes a fresh-work department which is one of the largest, if not the largest, of its kind in the world. The fourth floor contains the filler preparation and packing departments.

In these several departments the relative humidity varies from 62 per cent in the packing department to 70 per cent in the conditioning room. Temperatures are held automatically between 70 degrees and 78 degrees at all times, regardless of outdoor conditions or the season of the year. A brief description

Interior of One of the Battery of Ejector System Dryers, of Metal Panel Construction. In These Rooms the Stripped Tobacco Filler Is Dried in 36 Hours



Unit Air Conditioner Installed in the Finished Goods Seasoning or Storage Department. This Is a Complete, Self-Contained Air Conditioning System, Designed to Operate Independently of the Central Station System

of the air conditioning equipment and the functions of its various parts follows:

The Central Station System

One central station spray chamber or air conditioner, within which the air is washed and humidified or dehumidified according to the seasonal or daily weather demands, is located on the fourth floor and is the center of control for the conditions in the preparation, manufacturing and packing departments. Centrifugal fans draw the air through this air conditioner and distribute it through the metal duct systems to the various departments under control. The air issues from these ducts into the departments through carefully designed diffuser outlets so located on the ducts as to produce a uniform circulation of air throughout each room or department.

A Corner of the Packing Department. Controlled Atmospheric Conditions Are of Paramount Importance

Heaters are situated in the duct branches leading to individual departments, permitting individual control of heating, when desired, in these departments. Automatic thermostats control the dew-point or saturation temperature of the air at the spray chamber. Individual automatic temperature and humidity controls are located in each depart-

In the Sorting Department, Air Conditioning Plays the Important Part of Bringing the Tobacco Leaves to the Correct Condition Before Undergoing the Operations in Making the Cigar

ment and react upon dampers or steam valves to maintain the particular condition of temperature and relative humidity required in each department.

During the winter the system operates to cleanse the air of all dust and dirt drawn through from outdoors or from within the plant, to humidify the air, and to warm it. During the summer and, in fact, during all excessively humid seasons, the same system serves to reduce the humidity as well as to cool and cleanse the air before its distribution to the various departments.

For the purpose of cooling and dehumidifying, a single centrifugal refrigeration unit of 125 tons capacity operates in conjunction with the air conditioning system, supplying, under maximum conditions, approximately 500 gallons of water per minute, at approximately 52 deg. fahr., to the spray chamber. The refrigeration unit is located on the first floor in a room measuring 16 ft. by 32 ft., and occupies a floor space of but 100 square feet. Operation is by a 150 H.P. electric motor. A centrifugal pump circulates the water between the evaporator or cooler of the refrigeration machine and the spray chamber. Most engineers are now familiar with this centrifugal refrigeration unit, which utilizes a non-combustible liquid as a refrigerant, and operates throughout its entire cycle below atmospheric pressure. Under the conditions maintained in this particular installation the shell and tube evaporator or cooler is usually maintained at a vacuum of approximately twenty-five inches of mercury, while the shell and tube condenser operates at a vacuum of about seven inches of mercury, under the usual load conditions. The compact arrangement of this unit may be observed from the accompanying illustration.

The Unit Air Conditioner

A unit air conditioner is in use in the finished goods seasoning or storage department. This unit is entirely self-contained and delivers air directly into the room without a duct system. At such seasons as cooling and dehumidification are required, this unit receives a supply of cold water from the centrifugal refrigeration machine.

The capacity of the central station system is 68,650 cubic feet of conditioned air per minute, and that of the unit air conditioner is 2,500 cubic feet per minute.

Ejector Dryers

Aside from the general air conditioning equipment which is applied primarily for controlling the manufacturing operations and the conditions of material in the production departments, there is a battery of ejector system dryers within which conditioned air is circulated over trucks of cured tobacco to bring it initially to the desired moisture content at which it will be maintained and used throughout all succeeding operations. These rooms, with their automatic control and effective air circulation, have the advantage of establishing uniform drying and conditioning schedules at all seasons. The material treated in these roms is entirely uniform and the old factors of waste through over drying and crumbling or breakage are entirely eliminated. Decided savings in floor space and in labor of handling also have resulted in such factories through the application of conditioned air for the various drying and processing processes.

The installation of air conditioning equipment in the plant of Waitt & Bond is typical of the equipment installed in many of the finest tobacco factories throughout the United States; typical, also, of the equipment installed for dozens of other industries in which weather control is a vital factor.

And here let us point out again that we have chosen tobacco on account of its very nature and because of the almost complete dependence of machine production on dependable, uniform and controllable weather conditions. Thus has air conditioning become a highly valuable and often indispensable asset to the progressive manufacturer, freeing him from daily weather uncertainties, improving the quality of his product, contributing to the health, efficiency and happiness of his workers.

Present Development of the Heating Industry in France

By A. Beaurrienne, Paris, France

DEVELOPMENT of the heating industry in France has been greatly influenced by the economic conditions resulting from the War.

In large French towns, apartments consist chiefly of structures four to eight stories in height. Before the War, all new structures were provided with a central steam or hot water system, and progressively all old ones, in which the apartment rent exceeded two or three thousands francs a year, were provided with similar installations. The heating charge was apportioned among the tenants and was either included in the rent or put in as a separate charge. After the War, the cost of anthracite coal increased from the 1914 price of 45 or 50 francs a ton to 500 and 600 francs. Then special legislation was enacted which prevented landlords from increasing the rents to be paid by the tenants. This caused many difficulties.

In the apartments built before 1914, the law does not allow any increase of the rent above 100 per cent. So the income of landlords is out of balance with the maintenance expenses which are six times what they were formerly. Consequently, landlords refuse to improve old buildings. The result is that the tenants have been compelled to provide their own heat which often is done by means of a small boiler in the kitchen and 3 to 10 radiators in the rooms. These small boilers are certainly not economical, especially since coal can be bought only in small quantities on account of lack of storage space. In this way progress was retarded.

In new apartments, architects tried to avoid these

disadvantages by means of a plan in which each tenant has control over his individual heating expense. In this plan, a steam boiler is installed in the basement; steam is distributed by a vertical main to every apartment, then to radiators with horizontal piping; condensation is collected and measured by a meter, and is then returned to the boiler.

In other apartment buildings, there is a boiler for each apartment, condensation being measured by a meter. Hot water is distributed to radiators with the help of a pump or of compressed air.

This system is still a problem for the landlords. In one attempt at solution of the problem, a company has been created which takes the whole responsibility of heating. It buys fuel, employs the necessary labor and sells steam to tenants. Naturally, this company, when it is possible, uses one boiler room for several buildings. This plan however, often is hindered by the suspicion, the backwardness, and the demands of a few landlords. One way now being adopted to overcome these disadvantages is by means of a franchise for central station heating. Such a franchise was obtained about two years ago by the Compagnie Francaise de Chauffage Urbain. This concern has given over this franchise to the Compagnie Parisienne de Chauffage Urbain.

The latter company at first considered the installation of stations, about six miles from each other, and between the two a steam main 40 inches in diameter; but considering the great investment that would be necessary and the impossibility of receiving adequate returns for a long time, an agreement has been made between this company and the company that sells steam to tenants. The second company has increased its capital, the Compagnie Parisienne de Chauffage Urbain subscribing for a number of shares of this concern, the name of which is Compagnie Parisienne de Distribution de Chaleur.

The latter company is to erect local central stations with high pressure boilers. In these stations, heating steam will be expanded in turbines and electric current will be sold to an electric distributing company.

Use Both Mechanical and Natural Ventilation in Koppers Building

IN THE Koppers building, illustrated here, both mechanical and natural ventilation are used. The three basements below street level are all mechanically ventilated and heated by fresh air. Four exhaust fans which are direct-connected motor driven units, have a total capacity for the three basements of 69,250 c.f.m., and three fresh air fans for this area have a total capacity of 61,475 c.f.m. Fresh air and exhaust fans also serve the first, second and third floors, which will be occupied for the most part, by shops. The exhaust capacity for these floors is 17,700 c.f.m. and one fresh air fan supplies 26,950 c.f.m.

Mechanical ventilation is also provided in those areas where large numbers of employes are located, such as drafting room, the blueprint room, and the gymnasium, which is located on the thirty-second floor. Fresh air is supplied to these rooms by means of unit heaters and exhaust is by motor driven fans.

Ten exhaust fans for the building are located on the thirty-third floor and one exhaust fan operates in the service entrance on the first floor where trucks unload. In addition to the capacity noted for the basements and lower floors, the total capacity of the remaining exhaust fans for toilets, elevator machinery rooms, vaults and rentable area, amount to 110,250 c.f.m. Natural ventilation is used in rentable area which has been subdivided into comparatively small offices, where all of the windows have outside exposure.

No cooling equipment has been put in. Fresh air is drained into the building at the second floor level and is filtered by automatic oil filters and heated by steam coils at the fans in the third basement below street level.

The heating system installed is two-pipe vacuum, steam being purchased from a central heating plant. Upfeed risers are located on the exterior columns of the building, with room temperatures automatically controlled by thermostatic valves. This type of control also operates the steam supply to unit heaters.

KOPPERS BUILDING, PITTSBURGH, WHICH IS VENTILATED MECHANICALLY AND NATURALLY

The Design of Flanged Pipe Joints

By Sabin Crocker[1]

IN undertaking an exposition of the principles of design for flanged pipe joints, it is necessary to describe briefly some of the more common types of flanged joints. While this involves restating features which are already well known to many readers, such descriptions are required here as a basis for explaining the items of design which are the main purpose of this article. Among these items are methods for computing bolt stress due to internal pressure; bolt loading required to maintain a fluid tight joint; total tensile stress in bolts at root of threads; wrench torque required to produce a given tensile stress in a bolt; stresses induced in flange material through bolt loading and internal pressure.

Although little has been published regarding most of the above items, they actually are not difficult to co-ordinate and treat in a rational manner in flange design, and it is hoped that an explanation of how to go about it will be of assistance to many piping designers. The recent trend toward high temperatures and high pressures is responsible for the introduction of numerous special joints for severe service conditions. The variety of such joints renders it impracticable to work out the gasket compression, bolt loading, etc., required for all types, but these features have been computed for two radically different types which probably represent conditions of maximum and minimum bolt loading in power piping. Values for other types encountered can be determined readily by the methods explained in connection with the two types considered; i.e., raised face with gasket, and small tongue and groove.

TYPES OF FLANGED JOINTS

Several types of flanged joints used in power piping are shown in Figs. 1 to 8. Following is a brief explanation of the distinctive features of these types and their application to various classes of service:

The Plain Face Gasket Joint

Fig. 1 is the plain face gasket joint commonly used for low pressure water lines, compressed air, exhaust, building heating and the like where pressures do not exceed 125 lb. Flanges and fittings are cast iron. Rough machine bolts of commercial grade mild steel or wrought iron having square heads and hex nuts are used. Gaskets are usually red rubber or composition extending out to the bolts, although full face gaskets are sometimes used. With cast iron flanges it is not desirable to have the bolts located much beyond the contact surface if cracking flanges through excessive bending stress is to be avoided. For this reason a plain face is used in the 125-lb. American Flange Standard, and in the 250-lb. American Cast Iron Flange Standard the raised face extends practically out to the bolt holes. Companion flanges generally are attached to the pipe by threading as shown at (A), although Van Stoning is used as shown in (B), especially in the larger sizes. An outstanding advantage of Van Stone and similar loose flange joints is that the flanges can swivel on the pipe during erection, thus removing any difficulty about lining up bolt holes in adjacent flanges.

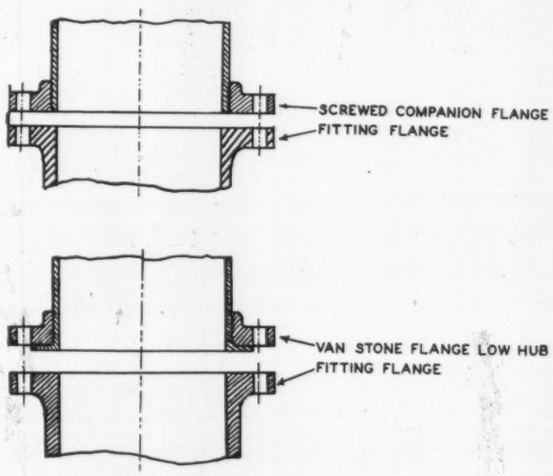


FIG. 1—PLAIN FACE GASKET JOINTS FOR LOW PRESSURES. 1A (ABOVE) SHOWS SCREWED COMPANION FLANGE AND FITTING FLANGE; 1B (BELOW) SHOWS VAN STONE FLANGE, LOW HUB, AND FITTING FLANGE

Raised Face Gasket Joint for 250-Lb.

Fig. 2 is the 1/16 inch raised face joint of the American Standard for 250 lb. in cast iron. In this case a high hub Van Stone flange is shown placed against a fitting. Screwed companion flanges are also used, particularly in the sizes 3-in. and smaller which are difficult to Van Stone.

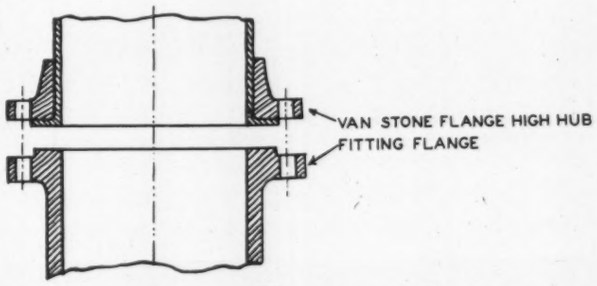


FIG. 2—RAISED FACE GASKET JOINT FOR 250 LB. IN CAST IRON.

Note 1: Engineer, Engineering Division of The Detroit Edison Company.

The raised face comes practically out to the inner edge of the bolt holes for the reason explained in the preceding paragraph. Semi-finished mild steel machine bolts with square heads and hex nuts are customary.

Raised Face Joint for 600-Lb. and Under

Fig. 3 shows a square-corner, full-thickness lap Van Stone flange placed against a raised face fitting flange of the American Steel Flange Standard. This

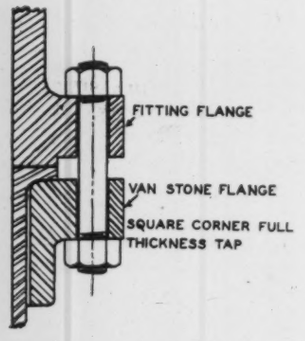
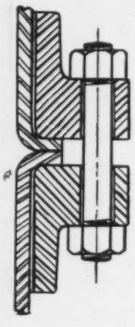


Fig. 3 (above)—Raised Face Gasket Joint, American Steel Flange Standards.

Fig. 4 (below)—Round Corner Van Stone Joint.



constitutes the standard raised face joint with gasket as commonly used for pressures below 600 lb. Alloy steel stud bolts with two hex nuts finished on the under side and with the upper corners chamfered are recommended. Metal or combination metal and asbestos gaskets are sometimes used, although long fibre asbestos gaskets are common practice. The lapped end of the pipe is upset in rolling so that the lap is the full thickness of the pipe wall after machining. The square corner is deemed advantageous in that it provides a smooth passage through the pipe bore and obviates pockets where condensation can collect. Tests have shown it to offer much greater resistance to pulling the pipe out of the flange than is offered by the round corner joint. With raised face joints it is desirable to machine the contact surfaces so as to obtain a serrated effect consisting of either spiral or concentric grooves which aid in gripping the gasket. Very satisfactory results are obtained with grooves about 1/64 inch deep and 32 to the inch. A serrated effect of this kind is sometimes termed "phonograph finish."

The Round Corner Van Stone Joint

Fig. 4 is the round corner Van Stone joint which is still used to some extent, but has been superseded largely by the square corner lap. The advocates of round corner style of Van Stoning claim that the metal in the lapped end is left in better condition than if it were upset to obtain the square corner. Equally satisfactory metallurgical results can be obtained in the square lap, however, if the metal temperatures are properly controlled during Van Stoning.

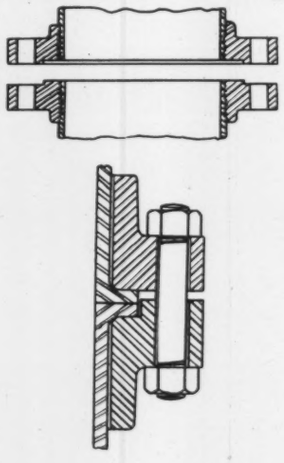
Male and Female Recessed Gasket Joint

Figs. 5A and B illustrate the male and female type of recessed gasket joint described in the American Steel Flange Standard. An advantage claimed for this and other types of recessed joints is that the overhanging shoulder tends to prevent blowing out of the gasket. Little trouble, however, is experienced on this score with the raised face joint shown in Fig. 3. A decided disadvantage in the case of fittings, or of the joint shown in Fig. 5A, is that the piping has to be moved axially in order to engage or disengage the joint. This is awkward in getting the last piece into a line, or in removing a piece after the line is once assembled. The loose flanges shown in Fig. 5B get around this difficulty.

Tongue and Groove Joint

Fig. 6A shows a pair of screwed companion flanges with the large tongue and groove described in the American Steel Flange Standard, while Fig. 6B shows a pair of Van Stone flanges with the small tongue and groove. This joint, as well as the male and female joint previously described, is applicable to fittings as well as to companion flanges. The groove or recess tends to retain the gasket and constrain its tendency to spread out under the high gasket compressions used with this joint. Gaskets are metal, usually copper, monel, wrought iron, or forged steel, sometimes copper-plated. Due to the narrow contact surface, it is possible to attain relatively high gasket compression without overstressing the bolts and flange material.

Fig. 5 (below)—Male and Female Recessed Gasket Joints. 5A (Above) Shows Screwed Companion Flanges; 5B Shows Van Stone Flanges



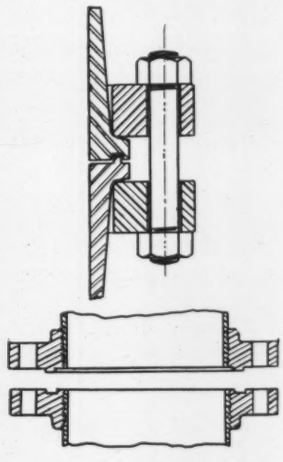


Fig. 6—(Above) Tongue and Groove Joints. 6A (at Bottom) Shows Large Tongue and Groove with Screwed Companion Flanges; 6B Shows Small Tongue and Groove with Van Stone Flanges

Loose Ring Joint

Fig. 7 is one type of loose ring joint of which there are a number in use. This joint somewhat resembles the tongue and groove described above, and generally has a forged steel ring which wedges tightly into the

grooves under bolt load. Loose ring joints are used extensively in oil refinery work and to some extent for high temperature, high pressure steam piping. Loose rings of rubber are used in some hydraulic lines, and can be kept tight, it is claimed, with very low compression of the rubber ring.

Sargol and Sarlun Joints

Fig. 8A is the Sargol joint which depends on a welded seal for fluid tightness, but carries all internal pressure and mechanical loads with the usual structure of flanges and bolts. This joint has been extensively used in high temperature steam plants operating at pressures from 600 to 1400 lb. The general dimensions are the same as the raised face joint of the American Steel Flange Standard.

Fig. 8B shows an improved type of Sargol joint known as the Sarlun.

FLUID TIGHTNESS OF PIPE JOINTS

Relation of Bolting to Internal Pressure: The primary purpose of a pipe joint is to connect pipe ends so as to obtain fluid tightness and requisite mechanical strength. Obviously the bolting must be more than adequate to prevent separation of the flanges by internal fluid pressure, usually considered as acting against an area circumscribed by the outer periphery of the gasket or other contact surface. To accomplish this end, the bolts must be given an initial tension exceeding the internal force tending to separate the flanges. The relation between total bolt tension and stress due to internal pressure only, is empirical and based largely on practical experience with different types of joints.

The American Steel Flange Standard states that, "The allowable working fibre stress considering internal allowable working pressure only, in bolting material for valve bonnet flanges, cleanout flanges, etc., shall not exceed 9,000 lb. per square inch, assuming the pressure to act upon an area circumscribed by the periphery of the outside of the contact surface." The bolting material referred to in this case is alloy steel having a tensile strength of from 95,000 to 125,000 lb. per square inch. No working stress for line flange bolts was stated by the A. S. A. Sectional Committee on Pipe Flanges and Fittings

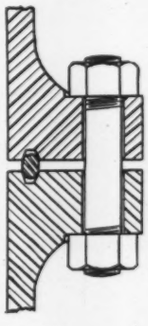


Fig. 7—Loose Ring Joint.

since its dimensional standards cover this feature. An analysis of these dimensional standards (See Tables I and II) indicates a tension in line flange bolts due to internal pressure only of from 1,100 to 7,000 lb. per square inch at the rated W. S. P. These are, of course, nominal values which did not take into account the additional bolt tension required to compress the gasket enough to hold it in place and maintain a fluid tight joint as explained in the next paragraph.

Bolt tension required to compress the gasket: The gasket compression required to maintain a fluid tight joint varies with the type of gasket, its width, and the means taken for holding it in place; viz., serrations on the contact surfaces, wedging in of the gasket by pinching it more tightly at its outer periphery, recessed construction, etc. For purposes of analysis, two radically different types have been selected as probably representing conditions of maximum and minimum bolt loading. It is believed that several common types can be assumed to approximate one or the other of the two types treated, or else considered as lying in the middle ground between. In any event, the methods of solution here used can be applied readily to other types of joints provided a basic assumption is made regarding the intensity of gasket compression required. Reliable data are scarce regarding the optimum gasket compression for any type joint or service conditions. While the gasket compressions stated in subsequent paragraphs may be higher than actually required, they represent conservative practice and probably come within 25 per cent plus or minus of meeting conditions existing in any properly made up joint.

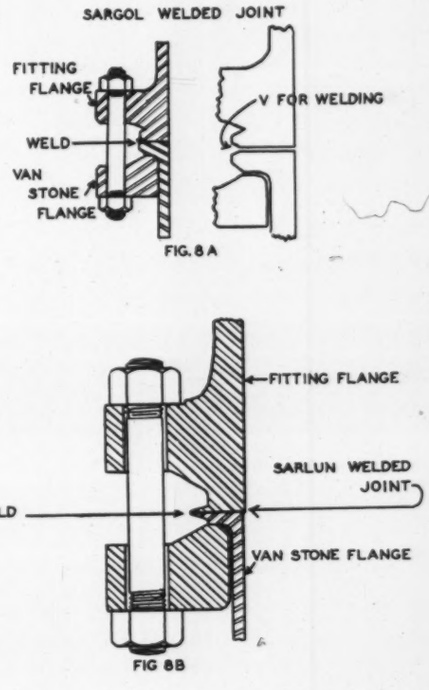


Fig. 8—A, Sargol Joint. B, Sarlun Joint

Raised face with gasket: The raised face flange with non-metallic gasket is the most common flanged joint used at the present time. For low pressure work the contact surface is usually given a smooth tool finish, while a serrated finish of some sort is used for higher pressures. The width of contact surface for each nominal pipe size and pressure rating is established in the dimensional standards of the A. S. A. In attempting to analyze bolt loadings and the bending stresses which they set up in flanges, some assumption is necessary regarding the intensity of gasket compression required to maintain a fluid tight joint. A rather general opinion exists among those who have worked with the design of the raised face joint, with a gasket compression[2] of at least 6 to 8

[2] "Gasket compression" as used here refers to intensity of compressive stress per unit area, measured in lb. per sq. in.

TABLE 1—BOLT LOADS AND WRENCH EFFORTS FOR RAISED FACE GASKET JOINT [1], AMERICAN STEEL FLANGE STANDARDS

Nominal Pipe Size	Bolt Tension, Lb. per Sq. Inch Corresponding to a Gasket Compression of 8 Times the W.S.P.					Bolt Tension, Lb. per Sq. In. Due to the W.S.P.					Total Bolt Tension Lb. per Sq. Inch (Due to Gasket Compression Plus the W.S.P.)					Wrench Torque Required Inch Lb. to Produce Total Bolt Tension at Left					Force Required Lb. at End of Usual Length Wrench				
W.S.P.	250	400	600	900	1,350	250	400	600	900	1,350	250	400	600	900	1,350	250	400	600	900	1,350	250	400	600	900	1350
½	.....	8,208	12,312	.....	.....	.....	1,180	1,775	.....	.....	.....	9,388	14,087	.....	.....	.....	120	181	.....	.....	...	17	26	...	....
¾	.....	7,107	10,661	.....	.....	.....	1,110	1,661	.....	.....	.....	8,217	12,322	.....	.....	.....	200	301	.....	.....	...	22	33	...	....
1	.....	9,328	13,992	10,894	16,340	.....	1,560	2,338	1,685	2,520	.....	10,880	16,330	12,579	18,860	.....	265	398	777	1,170	...	30	44	65	98
1¼	.....	14,582	21,874	16,766	25,164	.....	1,430	3,658	2,630	3,950	.....	16,012	25,532	19,396	29,114	.....	390	622	1,200	1,800	...	43	69	100	150
1½	.....	12,512	18,768	16,344	24,516	.....	2,150	3,220	2,650	3,980	.....	14,662	21,988	18,994	28,496	.....	586	878	1,730	2,600	...	53	80	124	185
2	8,860	14,176	21,264	16,200	24,300	1,600	2,550	3,840	2,770	4,150	10,460	16,726	25,104	18,970	28,450	255	408	612	1,172	1,760	28	45	68	98	147
2½	6,984	11,174	16,762	15,372	23,058	1,380	2,210	3,315	2,730	4,100	8,364	13,384	20,077	18,102	27,158	335	535	800	1,650	2,470	30	49	73	118	176
3	10,380	16,608	24,912	28,188	25,380	2,030	3,250	4,875	5,260	4,560	12,410	19,858	29,787	33,448	29,940	497	795	1,190	2,070	3,610	45	72	108	172	226
3½	11,684	14,466	20,198	24,235	29,851	2,460	2,830	4,240	4,850	5,790	14,144	17,296	24,438	29,085	35,641	566	1,070	1,510	2,650	4,300	51	89	126	189	269
4	14,454	16,656	24,984	23,530	29,700	3,110	3,570	5,360	4,640	5,440	17,564	20,226	30,344	28,170	35,140	703	1,250	1,875	3,400	5,400	64	104	156	212	300
5	18,480	21,293	24,389	23,515	25,942	4,350	5,000	5,720	5,080	5,050	22,830	26,293	30,109	28,595	30,992	913	1,630	2,740	4,400	7,210	83	136	196	244	328
6	15,718	18,080	20,712	25,409	26,568	3,920	4,500	5,150	5,850	5,530	19,638	22,590	25,862	31,259	32,098	786	1,400	2,350	3,780	6,140	71	117	168	236	307
8	15,234	18,611	21,989	22,752	26,374	4,300	5,370	6,110	5,760	5,840	19,534	22,580	28,099	28,512	32,214	1,210	2,050	3,400	5,460	8,980	101	147	213	273	390
10	11,174	13,526	17,141	22,514	26,352	3,620	4,390	5,150	6,220	6,250	14,794	23,981	22,291	28,734	32,602	1,330	2,895	3,430	5,500	12,500	95	181	191	275	462
12	10,940	14,717	18,826	25,452	26,892	3,780	4,750	5,700	6,880	6,450	14,720	19,467	24,526	32,332	33,342	1,780	2,995	3,775	6,180	12,750	111	166	210	309	472
14	9,554	12,429	16,042	22,932	.....	3,560	4,460	5,380	6,650	.....	13,114	16,889	21,422	29,582	.....	1,582	2,600	4,100	6,880	.....	99	145	205	313	....
16	9,280	12,758	17,712	24,617	.....	3,610	4,670	5,760	7,200	.....	12,890	17,428	23,472	31,817	.....	1,984	3,340	5,460	8,870	.....	110	167	248	385	....
18	10,730	13,792	18,941	24,214	.....	3,880	5,000	6,170	6,750	.....	14,610	18,792	25,111	30,964	.....	2,250	3,600	7,000	11,850	.....	125	180	304	438	....
20	9,534	12,883	18,326	23,774	.....	4,650	4,940	6,180	7,050	.....	14,184	17,823	24,506	30,824	.....	2,185	4,160	6,830	13,600	.....	121	189	297	485	....
24	8,770	12,205	17,659	19,894	.....	4,320	4,900	6,320	6,110	.....	13,090	17,105	23,979	29,374	.....	3,050	5,610	9,150	20,850	.....	138	225	339	578	....

Note [1]: Bolt tension refers to weakest section at root of threads; internal pressure considered as acting against circle circumscribed by outer edge of raised face, efficiency of contact surfaces assumed as 0.12, total bolt stress due to gasket compression plus W. S. P. assumed as identical to initial bolt tension set up with wrench; for standard dimensions of parts, see American Steel Flange Standard, for bolt areas, number of threads, etc., see Table 5, for further explanation, see text and sample problem.

times the rated working steam pressure is desirable in the steel flange standards when operating at temperatures up to 750 deg. fahr. This applies to conditions existing with pressure on the line, and a correspondingly higher gasket compression must be set up at the initial tightening of the bolts to compensate for the portion of the gasket compression which is relieved through the tendency for the bolts to stretch a bit when internal pressure is applied. The values given in Table I were computed for the Amer-

TABLE 2—BOLT LOADS AND WRENCH EFFORTS FOR SMALL TONGUE AND GROOVE JOINT [1], AMERICAN STEEL FLANGE STANDARDS

Nominal Pipe Size	Bolt Tension, Lb. per Sq. Inch Corresponding to a Gasket Compression of 16 Times the W.S.P.					Bolt Tension, Lb. per Sq. Inch Due to the W.S.P.					Total Bolt Tension, Lb. per Sq. Inch (Due to Gasket Compression Plus the W.S.P.)					Wrench Torque Required Inch Lb. to Produce Total Bolt Tension at Left					Force Required Lb., at End of Usual Length Wrench				
W.S.P.	250	400	600	900	1,350	250	400	600	900	1,350	250	400	600	900	1,350	250	400	600	900	1,350	250	400	600	900	1350
½	.....	8,883	13,300	.....	.....	.....	1,180	1,770	.....	.....	.....	10,063	15,070	.....	.....	.....	129	193	.....	.....	...	18	27	...	....
¾	.....	6,980	10,500	.....	.....	.....	1,110	1,660	.....	.....	.....	8,090	12,160	.....	.....	.....	197	296	.....	.....	...	22	33	...	....
1	.....	7,880	11,830	8,530	12,800	.....	1,366	2,050	1,480	2,220	.....	9,246	13,880	10,010	15,020	.....	225	349	620	928	...	25	38	52	78
1¼	.....	9,623	14,400	10,410	15,620	.....	1,970	2,950	2,130	3,190	.....	11,593	17,350	12,540	18,810	.....	283	423	776	1,162	...	31	47	65	98
1½	.....	7,217	10,820	8,910	14,520	.....	1,626	2,440	2,010	3,010	.....	8,843	13,260	10,920	17,530	.....	354	530	996	1,600	...	32	48	71	114
2	4,480	7,150	10,720	7,740	11,600	1,285	2,050	3,070	2,220	3,330	5,765	9,200	13,790	9,960	14,930	141	225	336	616	924	16	25	37	52	78
2½	3,475	5,560	8,350	6,870	10,300	1,145	1,830	2,745	2,260	3,380	4,620	7,390	11,095	9,130	13,680	185	300	444	830	1,245	17	27	40	59	89
3	4,365	6,745	10,400	11,200	9,680	1,740	2,780	4,140	4,500	3,880	6,105	9,525	14,540	15,700	13,560	244	381	582	970	1,640	22	34	53	81	101
3½	4,815	5,539	8,300	9,520	10,780	2,135	2,455	3,680	4,210	4,770	6,950	7,994	11,980	13,730	15,550	278	494	740	1,250	1,880	25	41	62	89	117
4	7,100	8,130	12,200	10,550	12,400	2,640	3,020	4,530	3,920	4,610	9,749	11,150	16,730	14,470	17,010	390	690	1,035	1,745	2,620	35	58	87	108	144
5	8,526	9,810	11,200	12,720	9,890	3,780	4,330	4,960	5,625	4,380	12,306	14,140	16,160	18,345	14,270	492	870	1,470	2,830	3,320	44	73	105	156	151
6	6,700	7,740	8,800	10,000	9,500	3,460	3,980	4,550	5,160	4,890	10,160	11,720	13,350	15,160	14,390	406	720	1,218	1,830	2,750	37	60	87	114	138
8	7,549	9,223	10,430	9,870	10,180	3,900	4,760	5,380	5,090	5,260	11,449	13,983	15,810	14,960	15,440	707	1,272	1,910	2,860	4,300	59	91	119	144	187
10	6,215	7,519	8,830	10,720	10,680	3,210	3,880	4,560	5,520	5,500	9,425	11,399	13,390	16,240	16,180	860	1,375	2,060	3,110	6,180	61	85	114	156	230
12	5,600	7,030	8,500	10,200	9,600	3,420	4,290	5,140	6,200	5,820	9,020	11,320	13,640	16,400	15,420	1,090	1,746	2,100	3,140	5,900	68	96	116	157	219
14	4,889	6,155	7,400	9,130	.....	3,230	4,050	4,900	6,040	.....	8,119	10,205	12,300	15,170	.....	980	1,574	2,360	3,530	.....	61	87	118	161	....
16	5,082	6,547	8,060	10,120	.....	3,270	4,225	5,190	6,530	.....	8,352	10,772	13,250	16,650	.....	1,290	2,060	3,080	4,640	.....	71	103	141	202	....
18	4,849	6,248	7,720	8,440	.....	3,565	4,600	5,675	6,200	.....	8,414	10,848	13,395	14,640	.....	1,298	2,080	3,730	5,600	.....	72	104	162	208	....
20	6,052	6,429	8,040	9,160	.....	4,260	4,520	5,650	6,440	.....	10,312	10,949	13,690	15,600	.....	1,590	2,550	3,810	6,850	.....	88	116	166	245	....
24	4,800	5,660	7,000	7,070	.....	4,010	4,550	5,840	5,670	.....	8,810	10,210	12,840	12,740	.....	2,050	3,340	4,910	9,050	.....	95	133	182	252	....

Note [1]: Bolt tension refers to weakest section at root of threads, internal pressure considered as acting against area of circle whose diameter is equal to the outside diameter of small tongue, efficiency of contact surfaces assumed as 0.12; total bolt stress due to gasket compression plus W. S. P. assumed as identical to initial bolt tension set up with wrench; for standard dimensions of parts, see American Steel Flange Standard; for bolt areas, number of threads, etc., see Table 5, for further explanation, see text and sample problem.

ican Steel Flange Standards for rated W.S.P. conditions and an assumed gasket compression of 8 times the W.S.P. This table is of particular interest in showing what high bolt tensions are required to meet the assumed conditions in the raised face joints.

Small tongue and groove: The small tongue and groove joint with metallic gasket probably requires as low bolt loadings as any type of gasket joint used for power piping. The loose ring type is in many ways similar to the small tongue and groove and probably requires about the same bolt loading. The gasket compression usually assumed for the small tongue and groove with pressure on the line is from 12 to 16 times the W.S.P. rating. The values given in Table II were computed for the American Steel Flange Standards for rated W.S.P. conditions and an assumed gasket compression of 16 times the W.S.P. Comparison with Table I indicates that much lower bolt loadings are required for the small tongue and groove joint than for the raised face. This is advantageous not only in lowering the tension required in bolt material, but it also results in correspondingly lower bending stresses in the flanges.

BOLT STRESSES

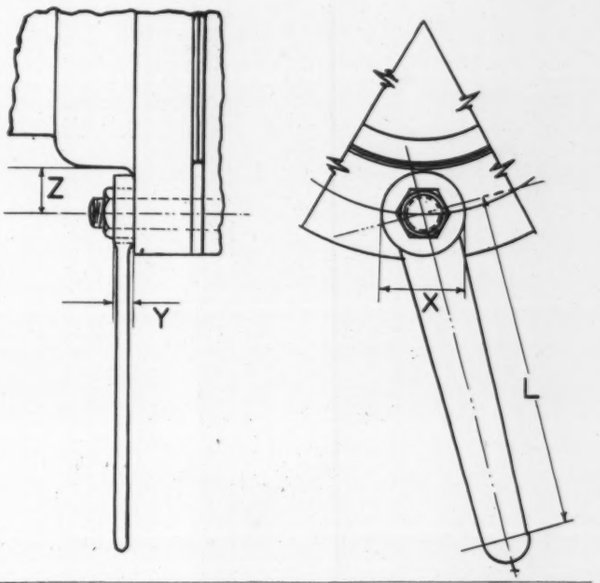
Initial bolt tension produced by tightening nuts: The tension set up in bolt material by tightening the nuts depends on (a) the pitch of the thread; (b) the torque applied with the wrench; (c) the efficiency of the nut and bolt thread in transferring wrench torque to direct tension in bolt material. These items are treated in detail below.

(a) In high pressure flange joints it is customary to use stud bolts with two hex nuts, made to U. S. standard dimensions and having the U. S. standard form of thread modified so that no bolt size has less than 8 threads per inch. The U. S. standard actually calls for fewer than eight threads per inch in sizes larger than one inch, as shown in Table III, but it has been found desirable in high pressure flanges to use 8 threads per inch in the larger sizes to secure a higher mechanical advantage in tightening the nuts. The eight thread per inch bolts also afford considerable gain in bolt strength as is shown by the column of effective area at root of threads and the column giving the ratios of these areas for different size bolts.

TABLE 3—COMPARISON OF U. S. STD. THREADS WITH MINIMUM OF 8 THREADS PER INCH

Diameter of Bolt Inches	Number of Threads per Inch		Area at Root of Threads Square Inches		Ratio of Effecting Areas at Root of Threads
	U. S. Std.	Steel Flange Std.	U. S. Std.	Steel Flange Std.	
½	13	13	0.126	0.126	1
⅝	11	11	0.202	0.202	1
¾	10	10	0.302	0.302	1
⅞	9	9	0.420	0.420	1
1	8	8	0.550	0.550	1
1⅛	7	8	0.693	0.729	0.95
1¼	7	8	0.890	0.930	0.96
1⅜	6	8	1.054	1.155	0.91
1½	6	8	1.294	1.405	0.92
1⅝	5½	8	1.515	1.681	0.90
1¾	5	8	1.744	1.980	0.88
1⅞	5	8	2.049	2.310	0.89
2	4½	8	2.300	2.655	0.87
2¼	4½	8	3.021	3.430	0.88
2½	4	8	3.716	4.290	0.87

NOTE[1]: See Table 5 for data used in computing these areas.



Bolt Diameter	X Diameter of Box	Y Thickness of Box	Z Note 2	$Z-\frac{X}{2}$ Note 3	L Effective Length
½	1½	25/64	13/16	1/16	7
⅝	1¾	31/64	15/16	1/16	9
¾	2 1/16	9/16	1⅛	3/32	11
⅞	2⅜	21/32	1¼	1/16	12
1	2⅝	¾	1⅜	1/16	14
1⅛	2⅞	13/16	1½	1/16	16
1¼	3¼	15/16	1¾	⅛	18
1⅜	3½	1	1⅞	⅛	20
1½	3¾	1 1/16	2	⅛	22
1⅝	4	1⅛	2⅛	⅛	23
1¾	4¼	1¼	2¼	⅛	25
2	4¾	1⅜	2½	⅛	28
2¼	5¼	1½	2¾	⅛	32
2½	5⅞	1⅝	3 1/16	⅛	36

Note [1]. All dimensions in inches.
Note [2]. Dimension "Z"= of bolt hole to foot of hub. Hub taper may vary but must allow full grip and free swing of wrench.
Note [3]. Minimum clearance between wrench and flange hub.

FIG. 9—USUAL DIMENSIONS OF BOX WRENCHES FOR U. S. STANDARD NUTS. (NOTE: NUTS OF U. S. STD. DIMENSIONS BUT HAVING NOT LESS THAN 8 THREADS PER INCH ARE RECOMMENDED FOR USE WITH AMERICAN STEEL FLANGE STD.)

(b) The average mechanic can exert a maximum pull on the end of a wrench of at least 165 lb. The actual pull in any case will be graduated to what he deems proper for the size bolt being tightened and probably will vary from about 50 lb. for a ½ inch bolt, to the maximum of 165 lb. for a 1 inch bolt. The length of wrench used increases with the diameter of bolt, usually running from about a 7 inch wrench (effective length) for a ½ inch bolt to possibly a 3 ft. wrench for a 2½ inch bolt as shown in Fig. 9. Referring to Table II for the tongue and

groove joint, it is evident that the necessary torque can be developed for practically all pipe sizes and pressure ratings by a mechanic using a usual length wrench. This is not the case for the larger sizes of pressure ratings above 250 lb. with the raised face joint, as is shown by Table I. In such cases it is customary to increase the length of wrench by slipping a piece of pipe over the handle, and if necessary have two or more men exert all the force they can at the end of the pipe. Another means of obtaining the required bolt tension is sledging the end of the wrench. Lubricating the threads is a considerable help in obtaining high bolt tension under these circumstances. Such measures should be used with considerable discretion, especially with bolt sizes smaller than one inch, or damage to the bolt or flange will ensue.

TABLE 4—METHOD OF COMPUTING WRENCH EFFORT FROM BOLT TENSION[1]

DIAMETER OF BOLT INCHES	NUMBER OF THREADS PER INCH	AREA AT ROOT OF THREAD SQ. INCHES	TORQUE IN INCH LB. AT CENTER OF BOLT	USUAL LENGTH WRENCH INCHES	FORCE REQUIRED AT END OF WRENCH LB.
d	N	a	T	L	F
½	13	0.126	0.0128t	7	0.00183t
⅝	11	0.202	0.0244t	9	0.00271t
¾	10	0.302	0.0400t	11	0.00364t
⅞	9	0.420	0.0618t	12	0.00515t
1	8	0.550	0.0911t	14	0.00650t
1⅛	8	0.729	0.1208t	16	0.00755t
1¼	8	0.930	0.1541t	18	0.00857t
1⅜	8	1.155	0.1914t	20	0.00957t
1½	8	1.405	0.2328t	22	0.0106t
1⅝	8	1.681	0.2785t	23	0.0121t
1¾	8	1.980	0.3281t	25	0.0131t
1⅞	8	2.310	0.3827t	27	0.0142t
2	8	2.655	0.4399t	28	0.0157t
2¼	8	3.430	0.5683t	32	0.0178t
2½	8	4.295	0.7116t	36	0.0198t

NOTE[1]: Based on relation (see text):

$$t = \frac{2\pi eNT}{a} = \frac{2\pi eNF1}{a}$$

from which

$$T = \frac{ta}{2\pi 0.12N} = \frac{ta}{0.754N}$$

and

$$F = \frac{ta}{2\pi 0.12N1} = \frac{ta}{0.754N1}$$

(c) The conversion of wrench torque into tension in the bolt material is aided through the mechanical advantage furnished by the inclined plane of the screw thread, but resisted in turn by friction in the threads and in the bearing surface between the under side of the nut and the flange. For well cut threads and machined bearing surfaces for the nut (i.e., the under side of the nut and the back of the flange machined or spot-faced) various authorities assign an efficiency of 11 to 12 per cent. If the threads and bearing surfaces are well lubricated, the efficiency can be increased to about 18 per cent. These values refer only to the efficiency of the contact surfaces as determined from the coefficient of friction, and do not involve the pitch of threads, which is taken into account as explained below.[3]

Formula relating wrench torque and bolt tension: The tension theoretically produced by wrench torque can be computed as follows, referring to Fig. 10, where F is the effort in lb. applied by a mechanic at the end of the wrench; T the torque in in-lb. produced by the wrench; L the length of wrench in inches; P the effort in lb. due to wrench torque at the mean thread circumference of the bolt[4]; W the resulting force in lb. which produces tensile stress in the bolt; d the mean thread diameter of bolt in inches; α the pitch angle of the thread; N number of threads per inch; e the efficiency as described in (c) above, but expressed as a decimal; a the cross-sectional area of the bolt at the root of thread in square inches; t the tensile stress in bolt material at root of threads, in lb. per square inch.

$$T = FL \qquad P = \frac{FL}{d/2} = \frac{2FL}{d} = \frac{2T}{d}$$

$$W = \frac{eP}{\tan\alpha} = \frac{2eFL}{d\tan\alpha} = \frac{2eT}{d\tan\alpha}$$

$$\text{where } \tan\alpha = \frac{\text{pitch of thread}}{\text{developed length of 1 thread}} = \frac{1}{\pi dN}$$

$$t = \frac{W}{a} = \frac{eP}{a\tan\alpha} = \frac{e}{a\tan\alpha} \times \frac{2FL}{d} = \frac{2eFL}{ad\tan\alpha} = \frac{2\pi eNFL}{a}$$

or

$$t = \frac{2\pi eNT}{a}$$

Example: It will be helpful to give an example of the use of the above formula at this time in connection with an explanation of how the values of Tables I and II are computed. The 1⅛ inch bolt of the 3½ inch pipe size of the 1350 lb. Standard having a raised face with gasket has been chosen for the example because it happens to involve the highest stress condition. Reference should be made to Table I in connection with the various items taken up.

Bolt Tension, Lb. per Sq. In. Corresponding to a Gasket Compression of 8 Times the W.S.P.: The gasket area in question is bounded by the outer circumference of the raised face and by the standard port opening of the fitting, as given in the American Steel Flange Standard. The diameter of raised face is 5½ inches and the corresponding area is 23.76 sq. in. The port opening is 3⅛ inches and the corresponding area 7.67 sq. in. The gasket area is 23.76 — 7.67 = 16.09. A gasket compression of 8 times the W.S.P. requires a force of $8 \times 16.09 \times 1350 = 173{,}772$ lb. The joint has eight 1⅛ inch bolts which, according to Table V, have a combined area at root of threads of

[3] For a discussion of the design of screw fastenings reference may be made to "Elements of Machine Design," by Kimball and Barr, 2nd Edition, Chapter XI. For the efficiency of bolt threads see also discussion by Oberhuber and Abele, page 767 of A. S. M. E. Transactions for 1925.

[4] The mean thread circumference is that corresponding to the mean thread diameter d and lies half way between the outer circumference of the bolt and the circumference bounding the area at root of threads. Since both P and d drop out of the final formula, it is not necessary to compute their numerical values.

TABLE 5—BOLTING FOR A. S. A. STEEL PIPE FLANGES

Nominal Bolt Diameter Inches	No. Threads per Inch	Major Diameter	Depth of Thread	Minor Diameter (at Root of Thread)	Area of Section at Root of Thread Square Inches	Number of Bolts in Flange Total Area of Section at Root of Threads, Square Inches					
						4	8	12	16	20	24
½	13	0.5000	0.04996	0.40008	0.126	0.504	.....	.....	.....	.....	.....
⅝	11	0.6250	0.05805	0.50690	0.202	0.808	1.616	.....	.....	.....	.....
¾	10	0.7500	0.06495	0.62010	0.302	1.208	2.416	3.624	.....	.....	.....
⅞	9	0.8750	0.07217	0.73066	0.420	1.680	3.360	5.040	.....	.....	.....
1	8	1.0000	0.08119	0.83762	0.551	2.204	4.408	6.612	8.816	.....	.....
1⅛	8	1.1250	0.08119	0.96262	0.729	.....	5.832	8.748	11.66	14.58	.....
1¼	8	1.2500	0.08119	0.08762	0.930	.....	7.440	11.16	14.88	18.60	22.32
1⅜	8	1.3750	0.08119	1.21262	1.155	.....	9.240	13.86	18.48	23.10	27.72
1½	8	1.5000	0.08119	1.33762	1.405	.....	11.24	16.86	22.48	28.10	33.72
1⅝	8	1.6250	0.08119	1.46262	1.681	.....	.....	20.17	26.89	33.62	40.34
1¾	8	1.7500	0.08119	1.58762	1.980	.....	.....	23.76	31.68	39.60	47.52
1⅞	8	1.8750	0.08119	1.71262	2.310	.....	.....	27.72	36.96	46.20	55.44
2	8	2.0000	0.08119	1.83762	2.655	.....	.....	.....	.....	53.10	63.72
2¼	8	2.2500	0.08119	2.08762	3.430	.....	.....	.....	.....	68.60	82.32
2½	8	2.5000	0.08119	2.33762	4.290	.....	.....	.....	.....	85.80	102.9

5.832 sq. in. The corresponding bolt tension is 173,772 ÷ 5.832 = 29,850 lb. per sq. in.

Bolt Tension, Lb. per Sq. In., Due to the W.S.P.: The area against which internal pressure acts is assumed as that bounded by the outer periphery of the raised face, or 23.71 sq. in. The corresponding force is 1350×23.71 = 32,076 lb. The corresponding bolt tension is 32,076 ÷ 5.832 = 5,790 lb. per sq. in.

Total Bolt Tension, Lb. per Sq. In., Due to Gasket Compression Plus the W.S.P.: This is the total of the two items above or 29,850+5,790 = 35,640 lb. per sq. in. Total bolt stress due to gasket compression plus W.S.P. has been assumed as identical to the initial bolt tension set up with the wrench. This assumption is explained as follows: If a gasket compression of 8 times the internal pressure is to obtain with pressure on the line, the initial bolt tension before pressure is applied must exceed that corresponding to 8 times the internal pressure, since the effect of applying pressure is to slightly separate the flanges, thus relieving part of the initial gasket compression. The amount of load relieved from the gasket corresponds to that exerted by the internal pressure, hence the initial bolt tension for the assumed conditions must be that due to a gasket compression of 8 times the W.S.P. plus that due to internal pressure.

Wrench Torque Required to Produce Total Bolt Tension: The total bolt tension in this case is 35,640 lb. per sq. in. The corresponding torque is determined from the formula $t = \frac{2\pi eNT}{a}$ by solving for T as follows (obtaining values of a and N from Table IV and using an efficiency e of 0.12 for dry surfaces):

$$T = \frac{ta}{2\pi eN} = \frac{35{,}640 \times 0.729}{2\pi \, 0.12 \times 8} = 4{,}300 \text{ inch lb.}$$

The same result can be obtained by the short cut method given in connection with Table IV as follows:

T = 0.1208t = 0.1208 × 35,640 = 4300 inch lb.

Force Required at End of Usual Length Wrench: Referring to Table IV, the usual length wrench for a 1⅛ inch bolt is 16 inches. Hence the force required at the end of the wrench is 4300 ÷ 16 = 269 lb. Or by the short cut method of Table IV,

F = 0.00755t = 0.00755 × 35,640 = 269 lb.

Torsional stress produced by wrench torque: In addition to producing tension in the bolt, wrench torque also produces a temporary torsional stress which is immediately relieved when the effort is removed from the wrench. The magnitude of this torsional stress can be computed from the fundamental torsion formula $T = \frac{qJ}{r}$, after making suitable deductions for torque absorbed in collar friction in the bearing surface between the under side of the nut and the flange, and for torque usefully converted into bolt tension. In other words, by considering that only the portion of the total wrench torque which is absorbed by friction of the thread surfaces tends to

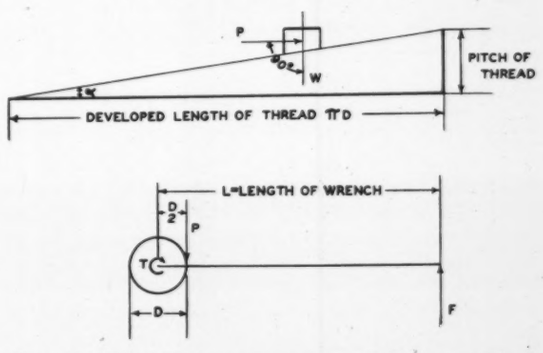


FIG. 10—BOLT TENSION PRODUCED BY WRENCH TORQUE.

create torsional stress in the bolt area at root of threads. It is convenient to assign subscripts to T to distinguish the division of torque contemplated, and define the other symbols as follows: T_w wrench torque, inch lb.; T_c torque absorbed in collar friction, inch lb..; T_t torque converted into bolt tension, inch lb.; T_s torque absorbed in screw thread friction, inch lb.; r one half the bolt diameter d at root of threads, inches; J the polar moment of inertia of the bolt section at root of threads, in inches to the fourth power; the coefficient of friction between the under

side of nut and back of flange (0.15 for dry surfaces and 0.10 for lubricated); W the load in lb. carried by the bolt; r_c the mean radius of the bearing surface on the under side of the nut.

Then $$T_w = T_c + T_t + T_s$$

from which $T_s = T_w - T_c - T_t$ (See numerical example below).

The equation for torsional stress in the bolt area at root of threads is

$$T_s = \frac{qJ}{r} \text{ or } q = \frac{T_s r}{J}$$

in which $J = \frac{\pi d^4}{32}$ for a circular cross section, and $r = \frac{d}{2}$, hence substituting,

$$q = \frac{T_s\ d/2}{\frac{\pi d^4}{32}} = \frac{5.1\ T_s}{\pi d^3} = \frac{16\ T_s}{d^3}$$

Since the magnitude of the torsional stress in a bolt is much smaller than the tensile stress t previously derived and does not give a combined stress much above that due to simple tension, it is usually neglected. It will be of interest, however, to compute the torsional stress and the equivalent combined stress for the case involving the highest bolt loading shown in Table I. This occurs in the 1⅛-inch bolts used with the 3½-inch pipe size of the 1350 lb. standard, which have a tensile stress t of 35,640 lb. per square inch and require a wrench torque T_w of 4,300 inch lb. It is first necessary to compute the numerical values of T_c and T_t to arrive at a value of T_s to use in solving for q. The computations are for dry surfaces.

The torque T_c absorbed in collar friction is computed from the formula $T_c = Wr_c$[5]. The coefficient of collar friction has been assumed as 0.15 for dry surfaces. W is the area of the bolt at root of threads times the tensile stress or $W = 35{,}640 \times 0.729 = 26{,}000$ lb. (See Table V for area at root of threads.) r_c is the radius from the center of the nut to a point lying midway between the root of the bolt thread and the periphery tangent to the hexagonal sides of the nut, or in other words

$$r_c = \frac{\text{(diameter of bolt at root of threads)} + \text{(distance across flats of nut)}}{4}$$

which in this case is, $r_c = \frac{0.9626 + 1.8125}{4} = 0.7$ inches,

approx. (See Table V for 0.9626 value and any table of U. S. Std. nut dimensions for 1.8125.) Then $T_c = 0.15 \times 26{,}000 \times 0.7 = 2{,}730$ in. lb.

The torque T_t usually converted to bolt tension is 12 per cent of the wrench torque T_w, since an efficiency of 12 per cent was used in computing the wrench torque for dry thread conditions. Hence $T_t = 0.12\ T_w = 0.12 \times 4300 = 516$ inch lb.

From which $T_s = 4300 - 2730 - 516 = 1054$ in lb.

and $q = \frac{5.1 \times 1054}{0.9626^3} = 6{,}040$ lb. per sq. in.

The maximum equivalent tensile stress t′ due to combined tension and shear can be computed as follows[6]:

$$t' = \frac{t}{2} + \sqrt{q^2 + (t/2)^2}$$

Applying this formula to the case above: $t = 35{,}640$; $q = 6{,}040$ and $t' = \frac{35{,}640}{2} + \sqrt{6{,}040^2 + \left(\frac{35{,}640}{2}\right)^2}$ $= 36{,}650$ lb. per sq. in. The equivalent tensile stress considering combined tension and shear is $36{,}650 - 35{,}640 = 1{,}010$ lb. per sq. in. higher than the simple tensile stress. This corresponds to an increase of under 3 per cent, which is actually less than the probable variation in coefficient of friction for screw threads and collar. Hence, in ordinary calculations, it appears safe to neglect torsional stresses in pipe flange bolts, especially since the additional stress is immediately relieved when wrench torque ceases.

[5] See Elements of Machine Design, by Kimball and Barr, 2nd Ed., page 277.

[6] See "Strength of Materials," by James E. Boyd, 2nd Edition, page 290.

Editor's Note: Part II of "The Design of Flanged Pipe Joints" will appear next month.

How to Weld Monel Metal Piping

Will you kindly describe in your columns the proper method for welding monel metal piping?

(Signed) A. S.

In order to be successful in welding monel metal, the welder should have a knowledge of the peculiarities of this metal as well as of its properties.

Monel metal is an alloy of nickel and copper containing about 67 per cent nickel, 28 per cent copper, with 5 per cent of other elements such as iron, manganese, silicon and carbon. In appearance, monel metal when new resembles untarnished nickel. It is used extensively in food handling machinery, textile dyeing machinery, photographic tanks, pickling tanks, marine construction and in chemical equipment.

In welding monel metal piping, a few simple precautions must be observed. First, a neutral flame should be used. The tip should be one or two sizes larger than would be required for steel of the same thickness. Cold drawn monel metal wire or strips cut from sheet, should be used as welding rod. As a general rule, flux is not required. The oxide film that forms on the surface of the puddle helps to protect the metal underneath from further oxidation. Keeping the outer envelope of the flame spread over the weld area will also aid in excluding air. The rod should be melted under this skin of oxide and slag. Any particles of dirt or foreign matter should be worked up into the slag by melting underneath them. Then, when the weld is built up well above the surface, as all monel metal welds should be, grinding will remove all oxide, slag and impurities, leaving only good sound metal in the weld.

A Simple Method of Determining Pipe Line Expansion

THE LINEAR expansion of any length of pipe is determined by calculating the difference between minimum surrounding temperature and the highest temperature of steam, gases, or liquids to be conveyed through the pipe.

Table 1 shows the expansion in 100 feet of pipe at various temperature differences. To arrive at expansion per foot, divide by 100 and then multiply by the number of feet of pipe for which the expansion is to be figured.

TABLE 1—INCHES OF LINEAR EXPANSION IN 100 FEET OF PIPE

TEMPERATURE DIFFERENCE DEGREES F.	WROUGHT IRON	STEEL	CAST IRON
0	.00	.00	.00
10	.08	.08	.05
20	.15	.15	.10
30	.23	.23	.15
40	.30	.30	.25
50	.40	.38	.36
100	.79	.76	.72
125	.97	.92	.88
150	1.21	1.15	1.10
175	1.41	1.34	1.28
200	1.65	1.57	1.50
225	1.87	1.78	1.70
250	2.09	1.99	1.90
275	2.36	2.26	2.15
300	2.58	2.47	2.35
325	2.86	2.73	2.60
350	3.08	2.94	2.80
375	3.46	3.31	3.15
400	3.63	3.46	3.30
425	4.05	3.86	3.68
450	4.28	4.08	3.89
475	4.62	4.41	4.20
500	4.90	4.67	4.45
525	5.22	4.99	4.75
550	5.55	5.30	5.05
600	6.26	5.98	5.70
650	7.05	6.71	6.40
700	7.86	7.50	7.15
750	8.75	8.36	7.96
800	9.76	9.31	8.87

Example: Wanted to know the amount of expansion in 60 feet of wrought iron pipe subjected to a maximum steam temperature of 353 degrees Fahr. and a minimum surrounding temperature of 53 degrees Fahr.

Calculations:

	Degrees
Maximum steam temperature.....	353
Minimum surrounding temperature...	53
Maximum temperature difference....	300

Expansion in 100 feet of wrought-iron pipe, when maximum temperature difference is 300 degrees Fahr. (from Table 1) equals 2.58 inches.

Expansion in 60 feet of pipe equals $\frac{2.58 \times 60}{100}$ equals 1.548 inches.

Table 2 gives properties of saturated steam, including data as to temperatures at various steam pressures, and is, therefore, of value in getting the maximum steam temperature,—information that is required for calculating the amount of pipe-line expansion.

TABLE 2—PROPERTIES OF SATURATED STEAM

GAGE PRESSURE	ABSOLUTE PRESSURE	TEMPERATURE DEGREES F.	CU. FT. VOLUME OF 1-LB. STEAM	WEIGHT OF 1 CU. FT. STEAM
Vacuum In. Mer.				
17.70	6	170.06	61.89	.01616
13.63	8	182.86	47.27	.02115
9.63	10	193.22	38.38	.02606
5.49	12	201.96	32.36	.03090
1.42	14	209.55	28.02	.03569
Pounds Pressure				
1.3	16	216.3	24.79	.04042
2.3	17	219.4	23.38	.04277
3.3	18	222.4	22.16	.04512
4.3	19	225.2	21.07	.04746
5.3	20	228.0	20.08	.04980
6.3	21	230.6	19.18	.05213
7.3	22	233.1	18.37	.05445
8.3	23	235.5	17.62	.0567
9.3	24	237.8	16.93	.05907
10.3	25	240.1	16.30	.0614
15.3	30	250.3	13.74	.0728
20.3	35	259.3	11.89	.0841
25.3	40	267.3	10.49	.0953
30.3	45	274.5	9.39	.1065
35.3	50	281.0	8.51	.1175
40.3	55	287.1	7.78	.1285
45.3	60	292.7	7.17	.1394
50.3	65	298.0	6.65	.1503
60.3	75	307.6	5.81	.1721
70.3	85	316.3	5.16	.1937
80.3	95	324.1	4.65	.2151
91.3	106	332.0	4.192	.2336
101.3	116	338.7	3.848	.2599
125.3	140	353.1	3.219	.3107
151.3	166	366.5	2.737	.3654
200.3	215	388.0	2.138	.468
255.3	270	407.9	1.718	.582
305.3	320	423.4	1.456	.687
355.3	370	437.2	1.264	.791
435.3	450	456.5	1.04	.96
535.3	550	477.3	.83	1.20

To find the temperature of superheated steam, take the temperature of saturated steam at comparable pressure and add to it the total degrees fahrenheit of superheat.

For example, 200 lb. pressure and 100 deg. superheat equals 388 deg. plus 100 deg. (superheat) equals 488 deg. total temperature.

Swedish Investigation of Heat Transmission Through Building Construction

By J. C. Peebles

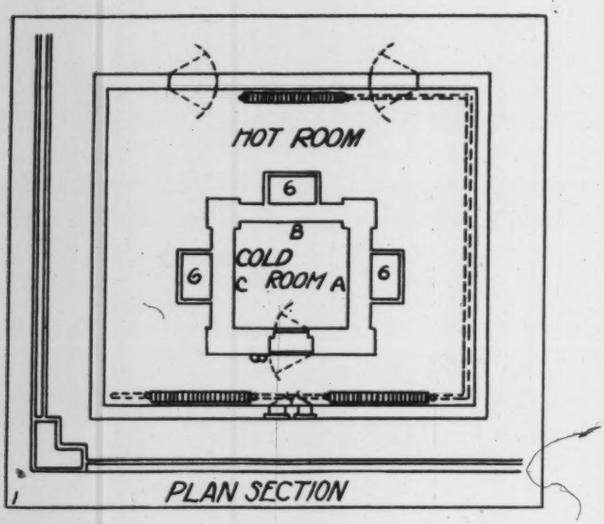


FIG. 1.—PLAN SECTION OF SWEDISH LABORATORY BUILDING AND TEST ROOM

THE INVESTIGATION referred to in this article is on heat transmission through building construction by H. Kreuger and A. Eriksson of the Academy of Engineering Science, Stockholm, Sweden. The books which record this investigation were made available to this publication through the kindness of P. Nicholls, U. S. Bureau of Mines, Pittsburgh, Pa. The original report was written in Swedish and translated as a basis for this article.

This is a field of industrial research which is receiving considerable attention in this country at the present time, and efforts are being made by the American Society of Heating and Ventilating Engineers, and by the committee on heat transmission of the National Research Council, to standardize and correlate the efforts of the many investigators who are engaged in this important work. It is interesting, therefore, to note what European engineers are doing in this field and to compare their methods with our own.

Nature's benefits are elusive; water leaks, food spoils, heat dissipates. The human race has devised many containersfor conserving human benefits, but probably the poorest container of them all is that heat container, —the human habitation.

A worthy effort directed toward making buildings better heat containers, is described in this article.

A special building seven meters (about 23 feet) square was provided in which to carry on the work. The walls were built heavily to simplify the problem of heating and temperature control. Heat was supplied by a hot water system supplemented by electric heaters. In the approximate center of the laboratory building a test room, 2.1 meters square, inside measurements, was erected.

One wall of this room is fixed in position while the other three are so made as to be removable. Three samples of the particular construction which is to be tested are used for the three removable walls of the test room, the entire arrangement being such that test walls can be removed easily and replaced with others. The room is cooled by means of refrigeration coils which are placed near the top in order to provide a natural air circulation.

In Fig. 1 is shown a plan section of the laboratory building and test room, while Fig. 2 shows a vertical section. The removable walls of the test or cold room are shown at A, B, and C respectively, in the plan section, and the refrigeration coils at 4 in the vertical section.

The Heat Flow Meter

The most important part of the equipment is the heat flow meter, shown at 6 in Fig. 1, by means of which the heat flow through the test walls is measured. It consists essentially of a box with one open side, mounted with this open side against the wall to be tested. The details of the box will be seen in Fig. 3, which shows a vertical section above and a plan section below. The walls of the box consist of two layers of wood with insulating felt between, to minimize heat loss and to facilitate temperature control. The open side of the box is one meter square, while the test wall, it will be recalled, is 2.1 meters sq. e. The test box is placed over the center of the test wall, a tight contact being secured by means of a rubber gasket.

Fig. 3 shows the heating coils, placed near the bottom of the test box, with two fans directly in front of them. Note that the fans are driven by a motor placed outside the test box, thus the heat developed by the motor does not affect the temperature inside the box. It will be noted also that a shield is placed between the fans and the test wall. This is to protect the wall from direct radiation from the heaters and also to facilitate control of the air circulation across the wall. The fans draw the air from the test wall, at the bottom, and deliver it over the heaters, thus providing a downward flow across the wall surface, similar to the conditions met in actual practice. The lower part of Fig. 3 shows plan section of test box.

Method of Testing

In conducting a test with this equipment, the air inside the test box is kept at the same temperature as that surrounding it in the test room. Thus there will be no flow of heat either to or from the test box, except that which flows through the test wall. Inasmuch as an accurate

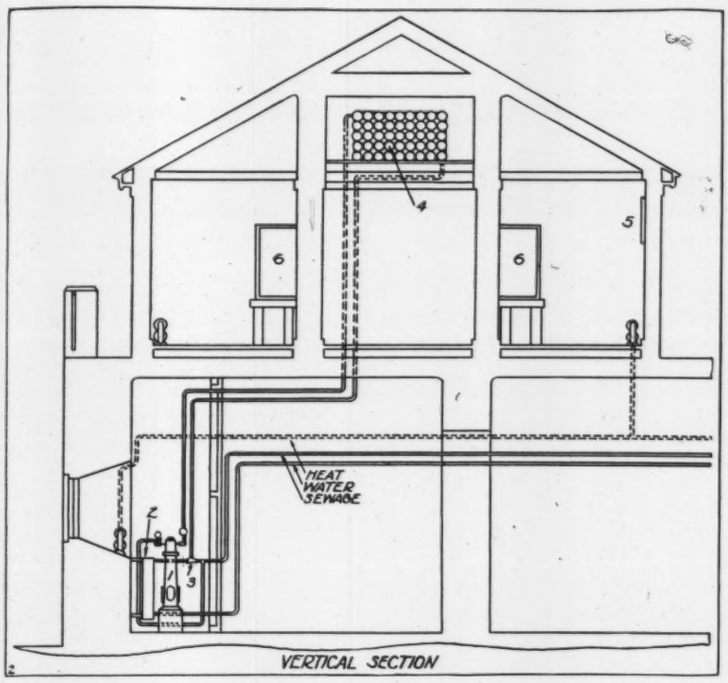


FIG. 2.—VERTICAL SECTION OF SWEDISH LABORATORY BUILDING AND TEST ROOM

measure of the electrical energy supplied to the heating coils is readily secured, the heat passing through the test wall is at once obtained. Direct current was used in coils, and voltmeter and ammeter readings were found to give the most accurate measure of the energy used.

From the above discussion, it will be evident that this scheme of testing requires the control and measurement of air temperatures in the cold room, in the test box, and in the test room proper. This is a task which every experimental engineer knows to be by no means easy; it is, therefore, entirely probable that the accuracy obtainable by this method is limited by this requirement. Since in this plan, the direction of heat flow through the test walls is inward towards the cold room, it follows that the natural flow of air will be downward across the outer surface and upward across the inner surface. Furthermore, the velocity of this flow will vary with the insulating merits of the wall under test, being very feeble for a good wall and much stronger for a poor one.

In the tests under discussion, natural air circulation was depended upon across the outer surface of the test wall, while inside the test box (but still on the outer surface of the wall), fan circulation was used as already indicated. The fans were run at such a speed as to give a temperature distribution at the wall surface inside the box just the same as at the same surface outside the box. (It will be recalled that the test box covers only about one quarter of the test wall surface.) This precaution insures a substantially uniform rate of heat flow through the test wall, the same in that portion lying under the test box as in the portion outside the box.

In the case of the cold room, fan circulation also was used in order to provide as nearly as possible, a uniform temperature distribution. Fig. 4 gives two sectional views of the cold room and shows the points at which temperature observations were made. Thermometer K_6 is placed in contact with the wall surface, with the others at varying distances therefrom as shown, the dimensions being given in centimeters. Thermometer K_1 is at the level of the top of the test wall and K_2 at the bottom.

A number of temperature surveys showed that the mean temperature occurred at a distance of approximately 1.25 meters (about 49 inches), from the floor of the room, at which elevation all thermometers except K_1 and K_2 were located. The circulating fan is shown in the plan section, located 1.25 meters from the test wall and delivering air thereto at an angle of 45 degrees. Note that the elevation of the fan is the same as that of the thermometers, 1.25 meters from the floor.

Fig. 5 shows the location of the thermometers used in the test room, on the warm side of the test wall. Five instruments are used to show the variations of temperature vertically, at a distance of 75 centimeters from the test wall. Four are used to give the horizontal variations, placed at an elevation of 1.25 meters from the floor and at distances from the wall as shown. The variations in temperature, both vertically and horizontally, depend only upon the thermal resistance of the wall, for no circulating fan is used in the test room.

As in the case of the cold room, the mean tempera-

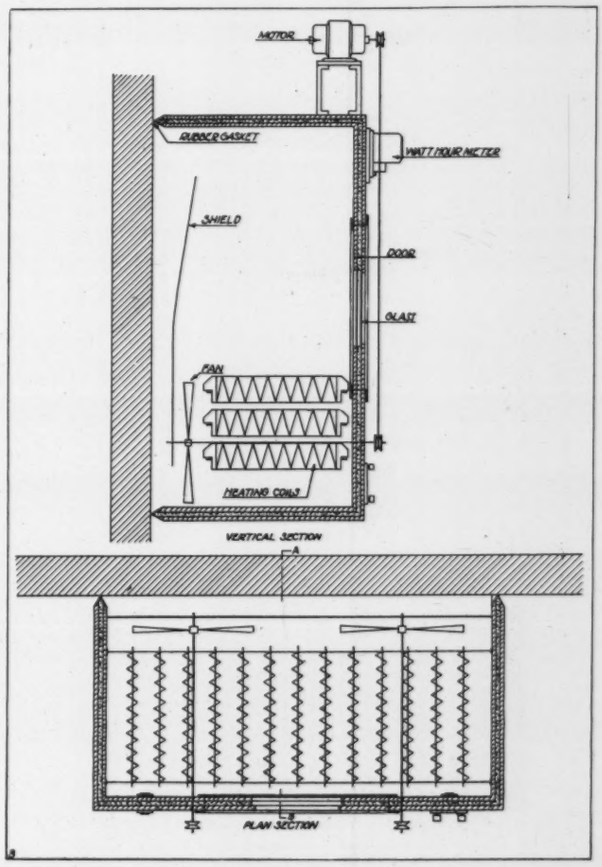


FIG. 3.—VERTICAL AND PLAN SECTION OF THE TEST BOX

ture of the air in the test room is found to be on a plane approximately 1.25 meters from the floor, and it is the temperatures at this elevation which are used to compute the heat transmission of the wall. Temperatures above and below this elevation are observed in order to show the maximum departure from the mean.

How Temperatures Were Measured

For measuring the temperatures inside the test box thermocouples were used, connected to galvanometers located in the test room. Air temperatures were observed at the same elevation as noted above for the test room, 1.25 meters above the floor. This is practically at the horizontal center line of the box, at which position the mean temperature was found. Additional thermocouples were mounted on the walls of the box so that equal temperatures could be maintained on both sides.

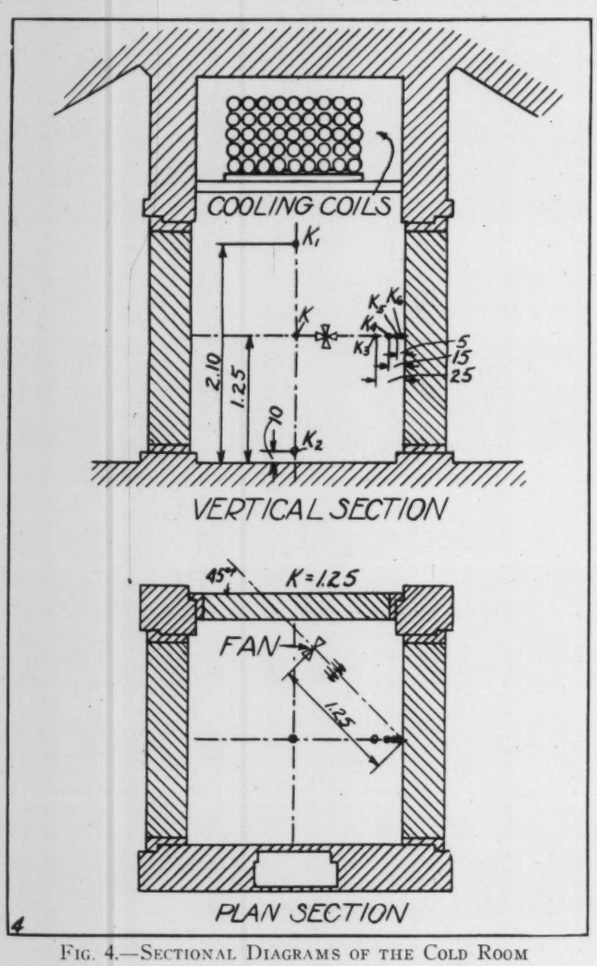


FIG. 4.—SECTIONAL DIAGRAMS OF THE COLD ROOM

If a double couple be used one junction can be mounted on the inside surface of the box wall and the other exactly opposite on the outside surface. Temperatures are then adjusted until the galvanometer reads zero, at which time the opposite surfaces are at the same temperature and hence no heat is passing between them.

In the cases where test-wall surface temperatures were measured with thermometers, a type of instrument having a flattened bulb was used. This bulb was placed in a shallow depression in the wall surface, the depth of which was half the thickness of the bulb, and the exposed surface covered with plaster of paris. The bulbs of thermometers used for measuring air temperatures were surrounded with paper shields to avoid radiation effects.

It was found, however, that the shields made practically no difference in the readings of the thermometers, the radiation effects being practically zero.

Method of Procedure

From the foregoing description of the equipment used by the Swedish engineers, the method of test will be readily apparent. After the test-wall has been placed in position as one side of the cold room, the refrigeration is turned on and the temperature brought down to about —18 degrees C., corresponding to approximately zero degrees Fahr. The test box or heat meter is then mounted in position with its open side against the outer surface of the test-wall. Thermometers and thermocouples as already discussed are placed in position, and electric current turned on in the test box heaters. The temperature of the test room is maintained constant and everything is operated steadily for twenty hours or more until uniform conditions are reached. The speed of the circulating fans in the test box must be regulated until the temperature distribution across the face of the test-wall within the box is the same as that without. Once established, all temperatures must be kept as nearly constant as possible, and complete observations made hourly over a period of at least eight hours. Test box and main test room are usually maintained near 18 degrees C., or 64 degrees F., although this may be varied as desired.

Swedish Equipment Similar to American Guarded Hot Box

American engineers familiar with heat transmission work will note at once the similarity between this Swedish equipment and the guarded-hot-box idea used in this country. The latter method makes use of a hot box almost identical with the test box of the Swedish method, but in place of our guard box surrounding the hot box, the Swedish engineers use the test room itself.

The difficulties inherent in maintaining constant temperature in air depend to a considerable extent upon the size and disposition of the space wherein the control is to be exercised. It is doubtless more difficult to maintain constant temperature conditions in a room seven meters square and say 2.5 to 3 meters high than in a box two meters square and 1.5 meters deep. It would seem that the most likely source of possible error in the Swedish method would be found in temperature differences between test box and test room, with a consequent flow of heat to or from the box.

In the American method, the test room is refrigerated and it is only necessary to maintain constant temperature conditions across the plane surface of the test-wall which forms one side of the test-box. This would seem to present fewer difficulties than to exercise such a control on five sides of a box projecting from the test-wall out into the test room.

The above objection can be minimized by making the walls of the test box quite thick and using material having very high thermal resistance. Also, by mounting numerous thermocouples or resistance thermometers on the walls of the test-box so that temperature differ-

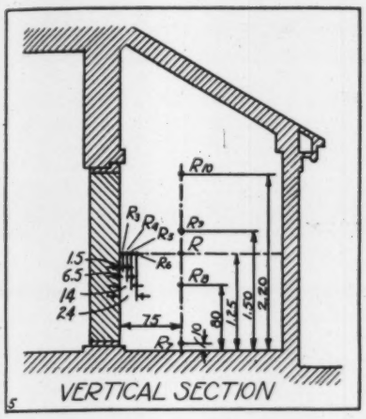


FIG. 5.—SHOWING LOCATION OF THERMOMETERS USED IN TEST ROOM ON WARM SIDE OF TEST WALL

ences readily may be discovered and corrected, less error is likely to occur. However, unless a very elaborate system of heaters is used in the test-box it will not be possible to compensate entirely for temperature differences between inside and outside, and hence this objection to the method can not be entirely removed.

On the other hand, the air circulation within the test-box, as provided by the Swedish method, gives a flow of air across that portion of the test wall covered by the box which is substantially equal to the flow provided by natural means on that portion of the wall not covered by the box. Thus air flow is practically uniform across the entire warm surface of the test wall, a condition very difficult to obtain with the American hot-box method. Inasmuch as air circulation has a pronounced effect on heat flow, it would seem that a method of test which takes this factor into account, in this respect at least, is superior.

It should be noted that the air circulation under consideration is not that due to wind, but simply the natural air movements across the face of a vertical wall which occur whenever a temperature difference between the two sides exists. Insofar as the Swedish method simulates this natural condition, it is apparently superior to our guarded-hot-box, in which, on account of the shape of the equipment, satisfactory control of air flow on the warm side of the test-wall is very difficult. If it is desired to investigate the effect of wind on heat transmission, both methods can be adapted to the purpose by providing increased fan circulation on the cold side of the test-wall.

It is probably sufficient for all practical purposes if the heat transmission of a wall under a given set of conditions can be measured with an error of 5 per cent or less. The method discussed herein should be capable of such accuracy, and with its use the Swedish engineers should be able to secure much valuable data. The equipment, when all necessary refinements are added, is elaborate and rather expensive, and tests will involve considerable time and labor. But they will be well worth while for there is much to be done in this important branch of industrial research.

Handling the Dust Problem in Lead Battery Manufacture

By H. L. Kauffman

IN THE plant of a large battery manufacturer a closed, dustless system of blending lead oxides and manufacturing battery paste has been installed during the past year. This system represents a marked advance in means for conveying, blending and mixing lead oxides. It accomplishes for this company an economy in manufacturing and provides a further insurance of safety to health of workmen.

Because the method of conveying and processing lead oxides in a closed system as used by this plant is applicable in its essential details to other plants having similar problems, the system there used will be de-

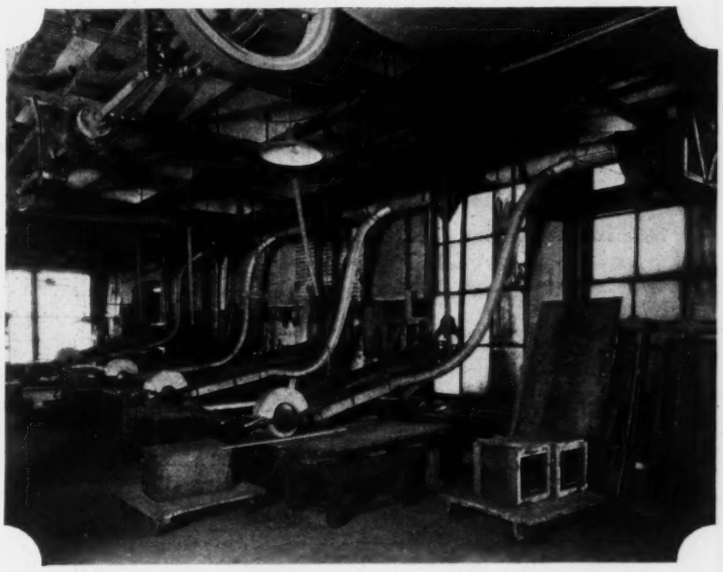


FIG. 1.—EXHAUST INSTALLATION FOR DUST FROM SWING GRINDERS

scribed in detail. In connection with the following description reference should be made to Fig. 4.

Wooden barrels *A*, containing oxides of lead, have six-inch metal heads which may be removed by a suitable tool. The suction nozzle *B* is inserted through this opening into the oxide which is conveyed by air to receiver *C* and from there discharged through a rotary discharge lock *D* by gravity to the storage bin *E*.

The capacity of the storage bin is four to six mixer charges, and two oxides of lead are transported and processed by the installation illustrated.

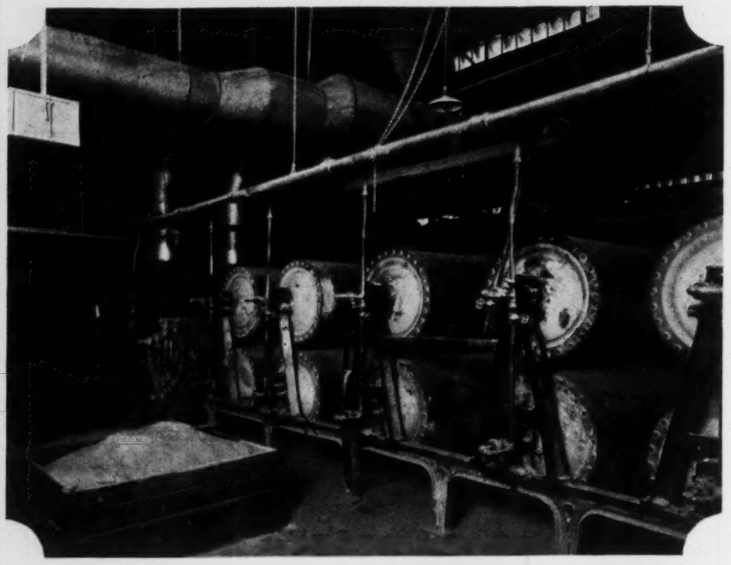


FIG. 2.—EXHAUSTING DUST FROM ROOFING PAPER MACHINERY

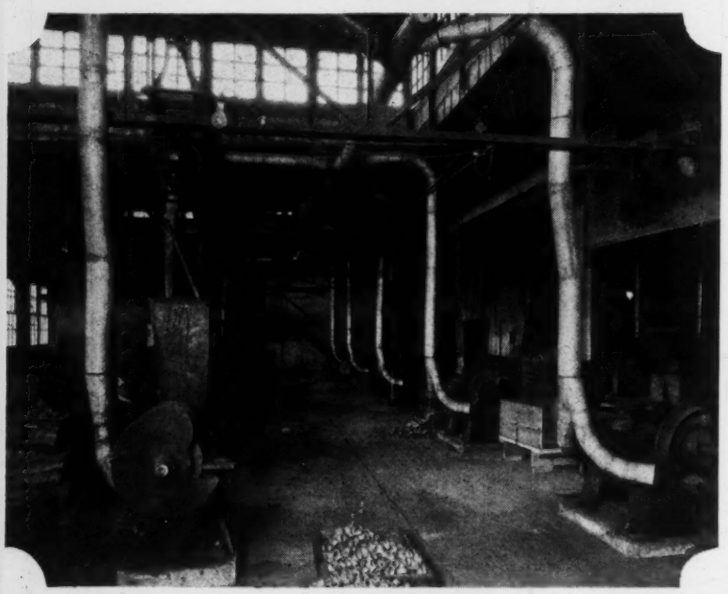


FIG. 3.—HOODS AND DUCTS IN DUST EXHAUST SYSTEM IN A MALLEABLE CASTINGS GRINDING ROOM

Conveying Oxides to Mixers

When the mixer *G* is to be charged, the oxide flows by gravity from the bin *E* through gates *H* into the weighing hopper *I*. The weighing hopper *I* is mounted on a scale *J*. When the desired amount of each material has been transferred to the weighing hopper, the control rod *K* is moved so as to open the control valve in the suction line which leads from the weighing hopper to the main receiver *L*, and at the same time closes the valve in the conveying line from the barrel being unloaded. Thus, the oxides are conveyed by suction from scale hopper *I* to the main receiver *L* and discharged through a rotary discharge lock *D-1* by gravity into any one of the four mixers *G*.

This is accomplished at will by means of a distributor *M* situated below the rotary discharge lock and regulated by a wheel installed on the same platform from which the weighing hopper is operated.

The air containing oxides of lead in suspension as a result of conveying them into receiver *L*, and also oxide dust emanating from conveying them from barrels *A* to receiver *C*, is drawn through filters *O*, thus

removing the fine lead oxide dust particles. The dust-free air leaving the filters is finally discharged to the outside atmosphere through the vacuum pump *P*.

The air-suspended lead oxides in the mixer, weighing hopper, bins and distributor are removed by a suitable filter *S*. The air enters the filters through an intake manifold *T* and the filtered air leaves through an outlet manifold *U* and finally to the outside atmosphere through a fan *V*.

The exit air passing through *V* and *P* is invisible; that is, if any dust is still present therein, such particles are smaller than the limit of visibility. As a matter of fact, for practical purposes, the recovery of lead-oxide dust by the filter is complete.

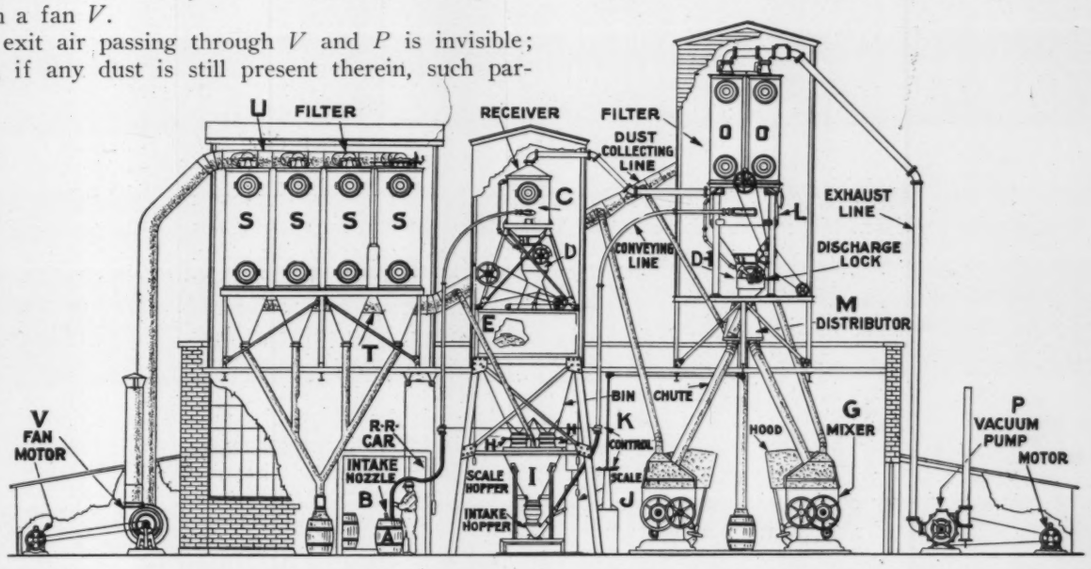


FIG. 4.—SCHEMATIC LAYOUT OF DUST-COLLECTING AND CONVEYING SYSTEM FOR LEAD-BATTERY MANUFACTURING PLANT

Use of a system for handling lead pigments as has been described prevents atmospheric pollution and insures to workmen freedom from health hazard, permits the recovery of a valuable product and lessens damage to mechanical equipment and appurtenances.

Due to advances in engineering technique, dust and fumes are no longer necessary concomitants to successful manufacturing.

Heat Utilization in District Heating

By Sterling S. Sanford*

In the district heating business, heat utilization refers to the economical use of heat by customers. It is important that steam be used with as little waste as possible so that the cost of heating with district steam will compare favorably with the cost of furnishing heat from the customer's own heating plant.

Utilization work starts with the architect's plans for new buildings to make sure buildings are designed so they can be heated at minimum cost. It is most important that it be possible to turn off steam in a building at night when it is not in use and that it be possible to turn off steam in parts of a building not in use at other times. It is also important that temperature control be provided. Water-heating economizers are usually justified.

All new customers are instructed in the economical use of steam. The operation of large buildings is followed during the first year or two. The work with other customers consists of following high-bill complaints, other buildings known to be excessive users of steam, customers on the demand rate, buildings in which the operating force has changed and those in which operating troubles have developed. The savings made are often very large. A large office building saved $2,900 in one month by turning off steam at night.

The most important points covered in investigating a building are:

1. Is steam shut off at night?
2. Is building zoned?
3. Is building overheated during day?
4. Are windows tight?
5. Is enough radiation installed?
6. Is heating equipment all in good condition?

Steam consumption records are kept for all buildings being investigated. These are corrected for temperature and compared with records for the previous year. New buildings are compared with others on the basis of steam used per cubic foot of space.

The effect of such work on the utility is:

1. Steam sold per square foot of radiation is decreased, thus decreasing revenue.
2. Night load on system is reduced and morning peak increased except in severe weather.
3. Diversity factor is increased.
4. Cost of work small part of total cost of service.
5. Customer good will increased.
6. New business easier to obtain.

If the district heating industry is to render the best possible service and at the lowest cost to the customer, it must undertake as much utilization work as possible. This will provide the foundation for future growth and usefulness.

* Central heating department, The Detroit Edison Company.

Heat and Energy Units

Fundamental Definitions

By Prof. W. R. Woolrich

IN THE mind of the average citizen, temperature and heat are the same. To him, a high temperature means an abundance of heat since the physical manifestations are so very evident.

To the heating engineer, it is very important that he should completely inform himself on the relation of heat and temperature.

Heat is a form of physical energy converted from some other form of energy, usually by friction, chemical action or by resistance to the transmission of light or electricity.

The transformation of any form of physical energy to heat energy is very easily done. Mechanical energy readily transforms to heat energy in the form of friction. Molecular or chemical energy readily transforms to heat energy as is evidenced by combustion of fuel or by the heat given up when water is mixed with sulphuric acid. Electrical energy is very easily transformed into heat as is demonstrated in the heating of motors, the heat losses in transformers and transmission lines, and the heating of commercial cooking equipment by electrical resistance units. The manifestation of heat transformed by light transmission resistance is best evidenced in the radiant heat experienced when a body is placed in the path of the radiant rays of the sun.

While the transformation of these various forms of energy to heat is very readily accomplished, to reverse this transformation cycle is usually more difficult. That is, the conversion of mechanical energy, chemical energy, electricity or light from heat energy is accomplished only with difficulty and considerable loss of available energy.

Heat may be considered, therefore, a very stable form of energy, since transformation to heat is readily accomplished, but the reversal of the cycle is obtained only with difficulty and available energy loss.

Temperature

Temperature is only a measure of the intensity of the heat and does not indicate the amount of heat. Thus, a small ladle of molten iron at 2400 deg. fahr. may contain much less heat than a heating boiler which is full of hot water at 200 deg. fahr.

Energy

Energy is the power or capacity of doing work. It is sometimes defined as accumulated capacity of mechanical work.

All kinds of energy can be reduced to terms of work although some forms are more readily convertible into mechanical work than others.

Energy that can be converted into work readily is usually called available energy. Examples of available energy are (a) dammed up streams that are converted into power through a water turbine (b) steam boilers under pressure available for engine or turbine operation, (c) volatile oils that can be used for operating gas engines, etc.

Energy that is not readily convertible into mechanical energy is usually called diffuse or unavailable energy. The principal example of a great reservoir of such energy is the molecular attraction forces within materials. For ages this form of energy has challenged the scientific world to make of it an available form of energy, but as yet no commercially successful process has been advanced.

Of the available forms of energy, further classifications are often desirable. These are those forms of energy that are available by virtue of (1) material position, and (2) speed, (3) chemical composition, (4) electrical or magnetic condition and by (5) temperature.

Energy that is available by virtue of its position is called potential energy.

Energy that is available by virtue of its speed is called kinetic energy.

These two forms require additional explanation.

Potential energy may be transformed into kinetic energy and vice versa.

The drop hammer in a forge shop has potential energy in its upper most position but just as it is about to hit or strike the metal at the bottom of the fall it is all kinetic energy.

Water passing over a precipice has mostly potential energy or energy of position at the top. At the bottom this energy is energy of speed or kinetic energy.

Steam under pressure behind a piston has a reserve of potential energy. As it rotates the fly wheel through the engine mechanism much of the potential energy becomes energy of speed or kinetic energy.

Dissipation of Energy

While energy may be transformed into less available forms it cannot be dissipated or destroyed. The statement, now commonplace, that "you can not get as much power out as you put in" is only relative. All of the "power put in" can be accounted for, but part of it has been converted into the heat of friction or some other less available form of energy. It is more correct, therefore, to state that the energy readily available is usually less after conversion has taken place.

The law of the conservation of energy states that

the sum total of energy in any system remains the same during any transformation.

Some very disturbing questions are often injected to try to disprove this law. For instance, "What happens to the energy put into compression of a metallic spring, if, after it is compressed, it is thrown into an active acid solution and completely dissolved by the acid?" By actual test, the energy used in the compression of the spring reappears in the increased heat of the chemical reaction of the metal. It can all be accounted for.

Units of Temperature

Temperatures between freezing and boiling may be measured by the sense of heat and cold conferred upon animal life. With the human being equipped with a temperature regulator maintaining his system at a fixed temperature when he is physically well, there is a very acute sensitiveness for temperatures to which his external body has become accustomed. He measures temperature on a scale of *hot* or *cold* from a cultivated sense based on his personal experience. Since this experience varies in individuals, hot or cold does not mean the same to each individual.

Temperatures are usually measured by some fixed values. In most engineering work in English speaking countries the Fahrenheit thermometer is used with the two fixed points on the scale; the melting point of ice and the boiling point of water at sea level. The former on the Fahrenheit scale is taken as 32 and the latter at 212.

For much of the scientific investigation work in the English speaking countries and for practically all temperature determinations in other than English speaking countries of Europe and America, the Centigrade thermometer is used. On this thermometer the temperature of melting ice is taken as the zero point, and the temperature of boiling water is taken as 100.

In some industries, a Reaumur scale is used with the melting point of ice taken as the zero point and the boiling point of water as 80.

Absolute Zero

Absolute zero on the Fahrenheit scale is –461. On the Centigrade scale it is –273. This zero of temperature is often defined as "the temperature at which a perfect engine would convert into mechanical work all of the heat supplied."

Sensible Heat

Heat may be present either as sensible heat or as latent heat. When changes in the heat present can be detected readily by one of the human senses such heat is called sensible.

Examples of this form of heat are most common. When water is placed over the stove we can feel it get hotter and hotter. Each additional amount of heat added gives a certain rise of temperature. Most of our common every day experiences are associated with sensible heat, and since we have no visible evidence of other heat being present we are most likely to assume that all heat is sensible heat.

Latent Heat

In the heating or cooling of many substances, heat is stored up due to internal changes that occur within the structure of the substance itself, yet no change of temperature may be experienced. Such heat is hidden or latent heat. Latent heat may be defined as heat added to a substance without changing the temperature of the substance.

For example, when we heat a kettle of water to the boiling point, it continues to get hotter and hotter until it reaches the boiling point. This heat is sensible heat. When the boiling point is reached no further change of temperature occurs within the kettle as long as the pressure remains unchanged. For the entire period of boiling away or vaporizing of the kettle of water after the boiling point is reached no increase of temperature is possible. Yet more and more heat is being added to the kettle to vaporize the water. The heat is being used to change the water from the form of water to the form of a vapor.

The amount of heat that has been added to change it from its liquid form to the vapor form of water is known as the latent heat of vaporization of water.

Unit of Measurement of Heat

The unit of measurement of heat is the British thermal unit. It is usually abbreviated as the B.t.u. The common definition for the British thermal unit is "The amount of heat necessary to raise one pound of water one degree Fahrenheit."

For example, if a pan contained 20 pounds of water and it is heated 10 degrees, the total number of B.t.u. added to the water will be 20×10, or 200.

A more exact definition for one B.t.u. is "The amount of heat necessary to raise the temperature of one pound of water from 62 deg. fahr. to 63 deg.

Another definition that has often been suggested is "1/180 of the amount of heat required to raise one pound of water from the condition of water at 32 deg. fahr. to the condition of water at 212 deg. fahr." By subtracting 32 deg. fahr from 212 deg. fahr. there is the difference of 180 deg. fahr.

Values of Latent Heat of Vaporization and Latent Heat of Fusion

The amount of heat necessary to raise one pound of water from water at 32 deg. fahr. to water at 212 deg. fahr. was given as 180 B.t.u. To vaporize this same pound of water at 212 deg. fahr. requires 971.7 B.t.u. This is the value of the latent heat of vaporization of water at atmospheric pressure. The 180 B.t.u. is in the form of sensible heat and the 971.7 B.t.u. is latent heat.

Likewise, when this same pound of water in the form of ice at 32 deg. fahr. is changed to water at 32 deg. fahr. it requires an addition of 143.4 B.t.u., though there is no temperature change. This is known as the latent heat of fusion of water.

Units of Energy

Since the power of doing work is called energy, the units for measurement of energy and work are the same. The most common unit of work is the foot

pound. It is defined as the amount of work necessary to raise one pound to a height of one foot.

By definition neither energy nor work involves time. Any length of time that might be required to raise the one pound weight to a height of one foot can be taken. When completed one foot-pound of work has been done.

When time is involved in the performance of this work, the product is power. Power is the amount of work done in a specified time, or, the capacity to do a piece of work in a specified time.

In mechanical and heating work the engine horsepower is defined as the doing of 550 foot pounds of work in one second or 33,000 foot pounds of work in one minute.

Relation Between Heat and Work

The commonly accepted mechanical equivalent of heat is 778 foot pounds = 1 B.t.u.

To express a relationship between heat and horsepower it is necessary to designate the period over which the power is measured.

Problem: Find the heat equivalent of 12 horsepower hours.

Computations:

1 horse power is defined as 33,000 foot-pounds per minute.

To maintain a rate of one horsepower for one hour it would be necessary to perform.

$60 \times 33{,}000 = 1{,}980{,}000$ foot-pounds of work in one hour since there are 60 minutes in one hour.

$\frac{1{,}980{,}000}{778} = 2545$ B.t.u. of heat in one hour. This is the heat equivalent of one horsepower hour.

Then the heat equivalent of 12 horsepower hours will be $2545 \times 12 = 30{,}540$ B.t.u.

Troubles from Obstructions in Steam Pipes

By C. W. Kimball

AN obstruction in large steam pipes is a rather rare occurrence and is generally not very difficult to find, but when the obstruction is only partial the trouble is a little more difficult to locate.

Some time ago, as the work on a new building was being finished, it was noticed that at times the north wing of the heating system bothered due to an occasional water hammer. The building consisted of three parts, a center section and two wings, piped exactly alike. The boilers were in the center building and the system was a plain two pipe gravity return system.

The water hammer occurred only at intervals and at different times during the week, sometimes not being noticed for several days. The piping system was examined carefully to see that the pipes were large enough, pitched correctly and that each part had a proper relation to the water line of the boilers.

A Pair of Overalls

The writer, after a series of experiments with a test gage, decided that there was a partial obstruction in the 4-in. main that supplied steam to the north wing. The heating contractor was skeptical, but after opening the pipes and pulling out a pair of overalls and a jumper was convinced that the writer's deductions were true.

The overalls and jumper apparently let enough steam through to supply the radiation, except when the rooms were cold, then the obstruction would act as a reducing valve and of course the water backed up in the returns due to loss of pressure in the main. By putting the test gage on each radiator it was quite easy to determine just where the obstruction was, as on the boiler side of the obstruction, the gage showed full pressure, while on the other side it showed a loss of approximately 1½ lbs.

No trouble was had with the north wing after the overalls and jumper were removed.

A Cement Bag

A somewhat similar trouble in connection with the heating system in a college building was found by the use of a test gage. The writer was called in after various attempts had been made, at an expense of more than $200, to remedy this trouble by changing the piping at various points.

The operating engineer had reached the conclusion that the whole system was defective and would have to be repiped to secure quiet operation.

After listening to all the stories, theories and deductions that the several investigators had advanced, the writer by using a gage found the trouble.

The system was a plain two pipe gravity return system operating at approximately two pounds pressure supplied through a reducing valve.

At times the piping would work perfectly quietly and properly and at other times water hammer would occur at various points. Careful testing at each radiator indicated some partial obstruction in the steam main and on taking the steam main apart at the point directed, a cement bag was discovered lodged in a 3 x 2½ x 2½ tee in this steam supply pipe. The bag was porous enough to let a certain amount of steam through, sufficient at times to supply the demand without trouble. At other times, when the total radiation was being supplied, the obstruction reduced the pressure enough to cause a backing up of the returns into the steam main, thus causing water hammer that annoyed the building occupants.

The removal of the cement bag made the entire system work properly and eliminated the annoyance that had existed for about a year.

Several Methods to Prevent or Correct Noises Made by Fans

ONE of the most important factors in public building and theater work is to keep the apparatus quiet and to prevent objectionable noise reaching the occupied portions of the building. In fact, industrial work is about the only class of ventilating in which noise is not a most serious and undesirable factor to encounter, and even then it is objectionable in a great many cases.

The first point to consider in the elimination of noise is the arrangement of the apparatus so as to operate within limits where the production of undue noise is not inherent with the installation. A distinction also must be made between "air-noise" and "machine-noise."

Production of Air Noise

Air noise is caused by moving air too rapidly and the noise made by the wind is a purely air noise. Machine noise is produced by the moving parts of a machine and is caused by friction in the bearings, gears, etc., the unbalancing of rotating parts, a reciprocating action of

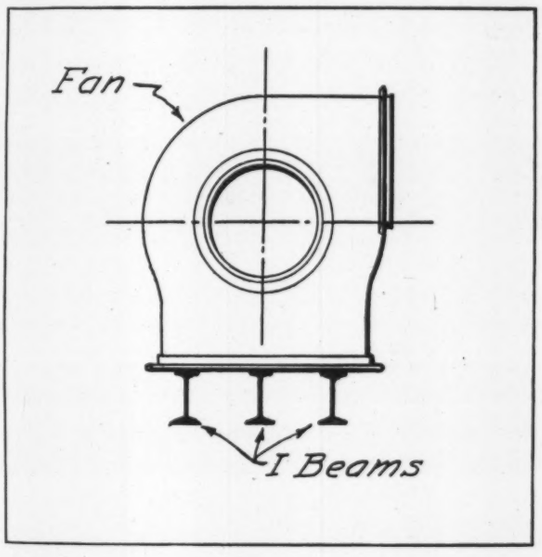


FIG. 1

various parts, and the striking of moving parts against fixed portions.

It is impossible, of course, to move either air or machinery without the production of any noise at all; but as long as the noise created is not of sufficient magnitude to cause annoyance outside of the apparatus room little more can be expected.

In medium and large size fans outlet velocities should be kept down to around 1600 to 1800 feet per minute and the tip speed to around 3200 or 3400 feet per minute. This, combined with duct and flue velocities of not over 1000 to 1200 feet per minute, will give an installation practically free of air noises.

Air noise is the simplest noise to eliminate in a ventilation system inasmuch as it depends on keeping the air velocity down to certain limiting speeds which can be provided for.

Production of Machinery Noise

Machinery noise is not as easy to control as air noise. This is because machinery noise may be produced by any one or a combination of several causes. Among the chief sources of machinery noise may be mentioned

(a) Too high speed
(b) Improper alignment
(c) Unbalancing
(d) Loose bearings
(e) Unstable supports

High speed is a relative term; what is high speed for a bicycle is not high speed for a railroad train and that which is high speed for a railroad train is greatly exceeded by the air ship. Thus, "high speed" is a relative term and cannot be said to exceed a certain given number of r.p.m. The chief objection to the so-called "high speed" is the fact that when apparatus is speeded up it generally results in aggravating any machinery noise

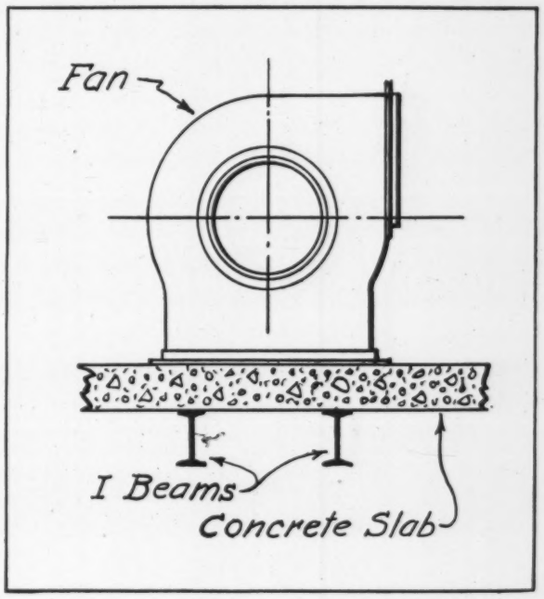


FIG. 2

which at a lower speed would be much less. This does not always happen but it usually does.

In cases where apparatus is not perfectly balanced or is out of alignment, increasing the speed of operation is

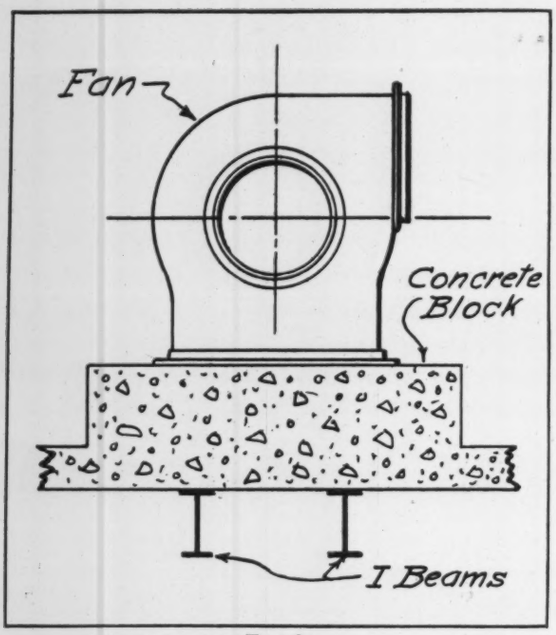


FIG. 3

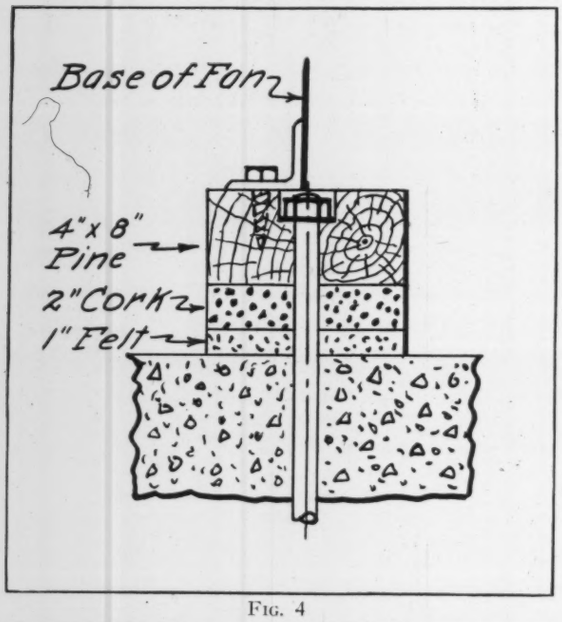


FIG. 4

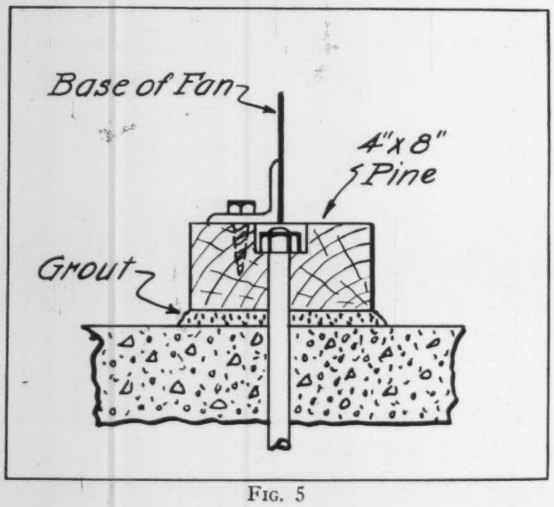


FIG. 5

bound to make matters worse, and a lessening of speed is likewise sure to make them better.

Sound Deadening Foundations

Vibration is often more objectionable than noise, and vibration itself can be the cause of added machinery noise. Consequently, if installations are made with the two-fold idea of preventing vibration and of stopping transmission of noise, a great deal may be accomplished even with noisy apparatus.

Taking the matter of vibration first, after securing

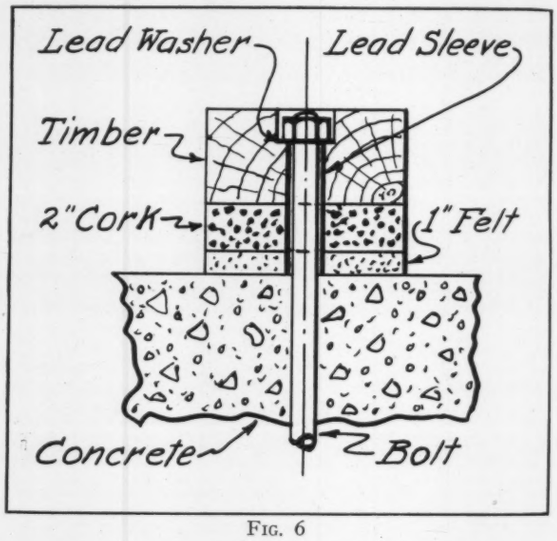


FIG. 6

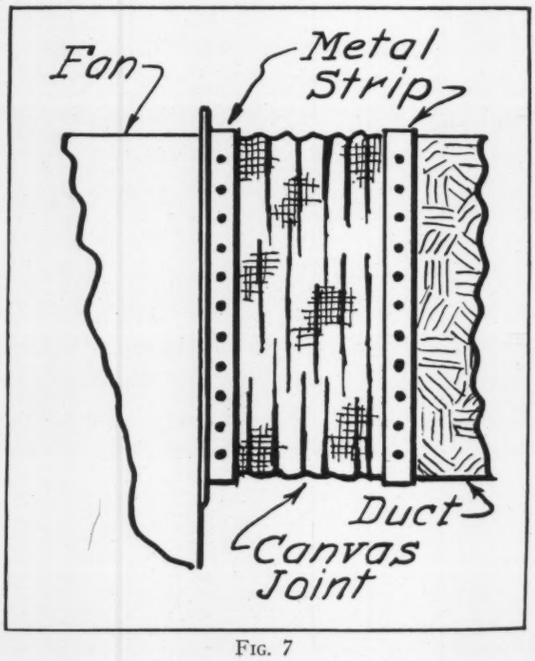


FIG. 7

the best alignment and most careful balancing possible a mass foundation should be used to absorb any and all vibration which still exists. The bare supporting of apparatus on structural steel of ample strength is usually very inadequate to provide sufficient mass to serve any useful purpose. (See Fig. 1). The addition of a 4-in. or 6-in. concrete floor slab on top of the steel is a help

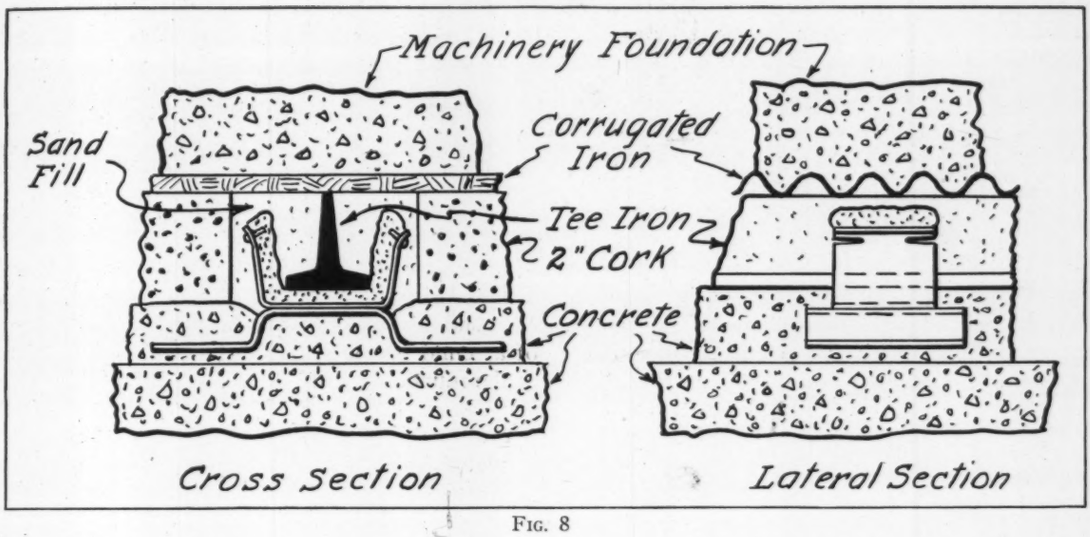


FIG. 8

(see Fig. 2) but the placing of a 12-in. to 18-in. thick block of concrete under the apparatus will provide a vibration absorbing capacity which will accomplish wonders. (See Fig. 3).

After sufficient mass has been supplied to absorb any reasonable amount of vibration the next point is to prevent what machinery sound there is from "telephoning" into the building structure. This is generally accomplished by insulating the fan from the foundation and from direct metallic duct connection.

In the case of the foundation, cork, felt, or both are used between the fan and the foundation. This is illustrated in Figs. 4 and 5. Lead is also used in bad cases, being usually placed directly under the fan base

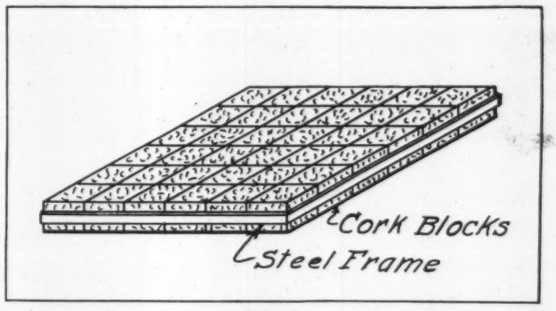


FIG. 9

in the shape of strips and under the heads of the lag screws in the shape of washers.

It is generally conceded that a satisfactory type of fan foundation consists of setting the fan on a 4-in. x 8-in. hard pine frame with halved joints which frame is placed over the foundation bolts and is countersunk to receive the nuts on the foundation bolts below the top of the frame.

The fan is lag screwed to this frame with strip lead under the bearing angles if desired. Between the frame and the floor is inserted 2 in. of cork (either with or without 1 in. of felt) and beneath this is placed the concrete mass to absorb the vibration. Then the frame is bolted to the foundation pad with lead sleeves around the bolts and lead washers under the nuts. This construction then appears as shown in Fig. 6.

Sheet Metal Connections

The foundation is not the only means, however, by which sound can be transmitted into the building or building construction. Nearly every ventilating fan has one or more sheet metal connections either to its inlet, its outlet or both. The metallic circuit of such connections can be broken by the insertion of a canvas joint in the duct work. The joints usually are made 6-in. to 8-in. long and are run around the outside of the duct with a metal reinforcing strip on the outside as shown in Fig. 7. The reinforcing strip is riveted to the duct work thus keeping the canvas from slipping off the metal pipe at either end of the joint.

Special Floor Construction

There is a type of patented floor which is now being used to a considerable extent for the purpose of killing machinery noise. This floor is constructed in various

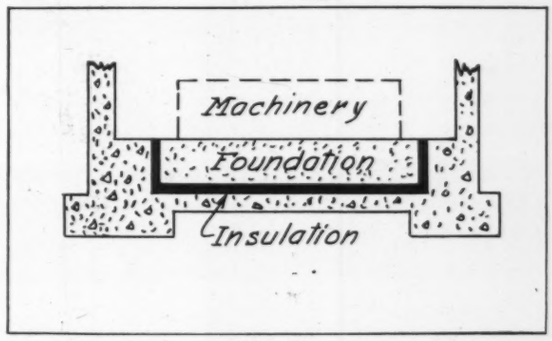


FIG. 10

ways but the principle followed out all the way through is to support the main members in felt padded saddles and the use of some type of sound deadening material between the main members. The saddles are arranged to carry either wood floor joints or steel tee irons and the space between is generally filled with cinder concrete or cork.

The method of building this type of sound proof floor is shown in Fig. 8 where the scheme of sound proofing is modified to accommodate a concrete floor with a concrete foundation under the apparatus. Here the chairs are cemented into a concrete bed and carry tee irons

in the felt covered arms. These tee irons have 2-in. cork board placed between with corrugated galvanized iron sheets laid over the tops of both the tee irons and cork.

Around the tee irons a filling of sand is poured and a solid flush surface is thus obtained on which to lay the corrugated iron. Then the concrete machinery foundation is poured on the top of the galvanized iron.

Modifications when applied to wood frame floors consist of substituting wood for the corrugated iron and setting the machinery on a heavy frame resting directly on the wood. Cinder concrete is also used in some instances in place of cork board.

Where machinery foundations come into contact with building foundations there always is the likelihood that vibration will be transmitted into the building. The first thing to do in such a case is to put in enough "mass" to absorb as much of this vibration as possible and then to insulate the "mass" from the building structure so that any vibration which is not absorbed will be prevented from being transmitted into the building.

One of the expedients often used for this purpose is an insulating mat made up of small squares of cork in a checker board arrangement and held in place by being arranged tightly inside of a steel frame made of flat steel in the form of a hollow square. This will be understood more clearly by referring to Fig. 9.

These groups or clusters of cork blocks are then used between the machinery foundations on bottom, sides, and both ends, as illustrated in the typical example shown in Fig. 10. It will be seen that there can be no direct connection between the foundation of the building and the foundation of the machinery when arranged in this manner and some very good results have been obtained.

It should be noted that insulation under the bottom of a foundation will not be sufficient if the ends or sides of the foundation establish a direct contact with the building construction. The best results are obtained when not only the bottom but also the sides and ends are insulated. Even carrying a cement floor over the top of the insulation so that it comes into contact with the machinery foundation is liable to cause trouble. To kill noise and vibration the insulating must not only be done thoroughly but completely; half way measures are largely a waste of money.

Control of Dust from Column and Post Sanders

CONSIDERABLE interest has been shown in the investigations of the New York State Department of Labor in refuse removal and dust control necessary in the operation of woodworking machinery, but specific details relating to many of the machines have been dealt with only in a limited degree.

It is stated by the department that machines, in connection with which sandpaper or other abrasive paper is used, when operated, create and disengage into workroom air, dust of a finely divided nature of both organic

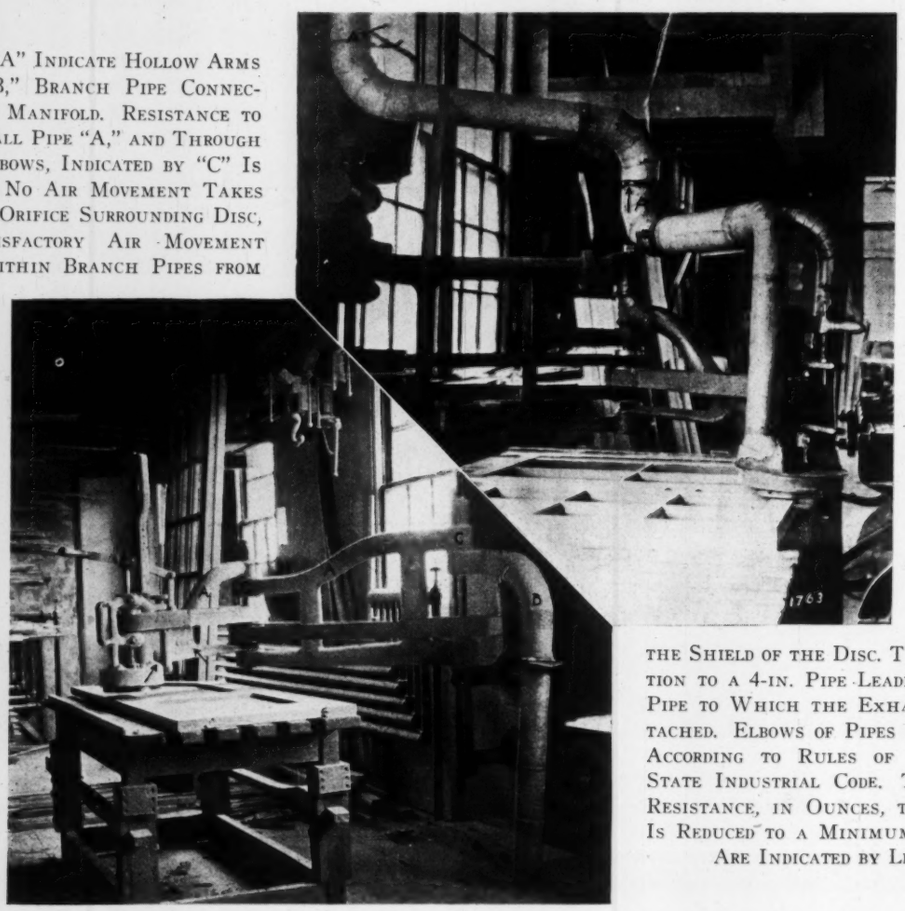


FIG. 1—LETTERS "A" INDICATE HOLLOW ARMS OF MACHINE; "B," BRANCH PIPE CONNECTION LEADING TO MANIFOLD. RESISTANCE TO AIR FLOW IN SMALL PIPE "A," AND THROUGH RIGHT ANGLE ELBOWS, INDICATED BY "C" IS SO GREAT THAT NO AIR MOVEMENT TAKES PLACE THROUGH ORIFICE SURROUNDING DISC, ALTHOUGH SATISFACTORY AIR MOVEMENT TAKES PLACE WITHIN BRANCH PIPES FROM OTHER MACHINES ATTACHED TO SAME EXHAUST SYSTEM. NOTE DEPOSIT OF DUST PARTS MACHINE INDICATED BY ARROWS, DUE TO LACK OF PROPER AIR MOVEMENT THROUGH ORIFICE

FIG. 2—COLUMN SANDER PROVIDED WITH TWO 3-IN. PIPE CONNECTIONS ATTACHED TO THE SHIELD OF THE DISC. THESE PIPES JUNCTION TO A 4-IN. PIPE LEADING TO THE MAIN PIPE TO WHICH THE EXHAUST FAN IS ATTACHED. ELBOWS OF PIPES ARE CONSTRUCTED ACCORDING TO RULES OF THE NEW YORK STATE INDUSTRIAL CODE. THE THEORETICAL RESISTANCE, IN OUNCES, TO THE AIR FLOW IS REDUCED TO A MINIMUM. SWIVEL JOINTS ARE INDICATED BY LETTERS "A"

and inorganic composition. The removal of dust from such machines is necessary by the use of effective exhaust systems, in order to prevent the operators from inhaling it.

The column or post sander in connection with which abrasive paper is used, is composed of two swinging arms on which pulleys and belts are operated to revolve a disc on which the abrasive paper is fastened. The disc is protected by a cast iron or steel shield to which the exhaust pipes are fastened for the removal of the dust created by its use. The machines are used for sandpapering doors, sash, blinds or any plane surface, and are constructed to operate over a surface equal to the radius which the arms will inscribe when fully extended.

As the point of operation of so-called "sandpapering" may be changed at the will of the operator, means to remove the dust must be employed at the point of origin, or before it moves ½-in. beyond the periphery of the revolving disc.

It has been customary in connection with these machines to exhaust air through the orifice formed between the apron which extends downward from the cover and the periphery of the disc. This is done by attaching an exhaust pipe to the disc's cover and using the hollow arms of the machines as a conveying pipe. At the end of this, a connection is made with a branch pipe leading to the manifold and exhaust fan.

As the hollow arms of the machine have a small internal diameter of approximately 1½-in. equaling an area of 1.76 sq. in. and the air must turn four or five right angles in its travel from the disc to the branch pipe attachment, a high resistance is created to the moving air. Fig. 1 depicts a column sander in connection with which the hollow arms are used as described.

The effectiveness of the exhaust system, in connection with the dust removal from the machine is dependent upon the velocity of the air through the orifice between the apron and the periphery of the disc.

This air stream must be of sufficient power to change the direction of the moving abrased particles of wood and abrasive to a vertical direction before they move ½ inch beyond the periphery of the disc. This may be accomplished by maintaining a minimum air movement of 4,005 feet per minute through the orifice.

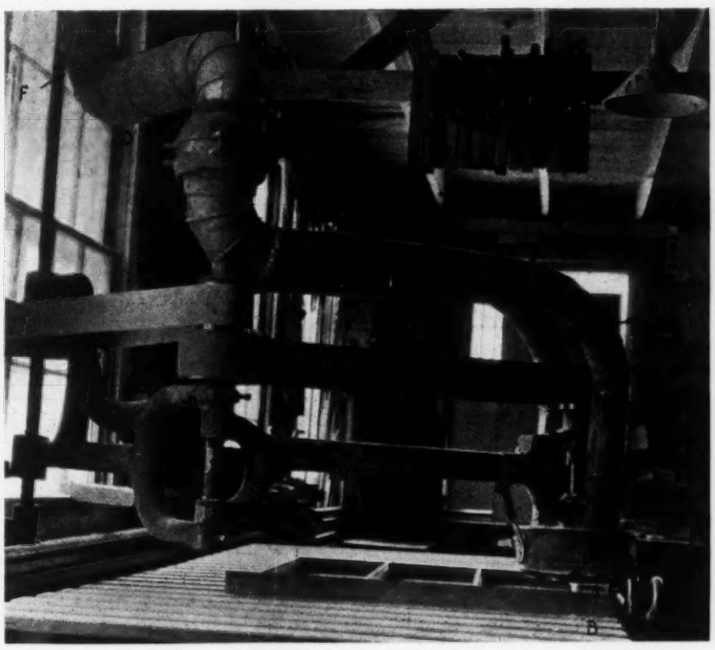


FIG. 3—THE RESISTANCE TO THE AIR FLOW IN THE PIPING AS USED IN THE ABOVE ILLUSTRATION, IN WHICH THE SAME AIR VELOCITY, 4005 FEET PER MINUTE, IS USED, IS THE SAME. LETTER "A" INDICATES 3-IN. JUNCTION PIPES; "B" ORIFICE BETWEEN SHIELD AND DISC; "C" HOLLOW ARMS OF MACHINES; "D" BALL AND SOCKET JOINTS; "E" SHIELD; "F" ELBOW OF 4-IN. PIPE CONSTRUCTED WITH RADIUS IN THROAT TO OFFER LEAST POSSIBLE RESISTANCE TO AIR FLOW

The circular orifice, formed between the 8-inch diameter disc and the 9-inch diameter apron of the shield possesses an approximate area of 13 square inches. Therefore, if a branch pipe of 4 inches diameter is used, having an area of 12.5 square inches, the respective areas would approximately coincide.

By the substitution of the 4-in. diameter pipe for the hollow arms, a conduit is provided in which the friction to the air flow, at 4,005 feet per minute, is reduced from 8 ounces created in the use of the hollow arms to 3¼ ounces in the 4-inch pipe. Besides, seven times the volume of air is moved through the orifice by the use of the 4-inch pipe.

The joints of the piping may either be swiveled or of the ball and socket type. Three-inch branch pipes attached to the disc's shield successfully may be used, joining them together to form the 4-in. pipe.

In connection with exhaust for machines depicted in Figs. 2 and 3, a higher air movement than 4,005 feet per minute was maintained in the piping,—approximately 5,000 feet.

Low Humidity Drying

By Malcolm C. W. Tomlinson

Article No. 2

SINCE human comfort is concerned mainly with temperature and relative humidity it is best, for those who are not so familiar with psychrometry, that an attempt be made to define these factors before we discuss comfort.

The atmosphere around us is known as air. Actually it consists of a varying mixture of water vapor and air. When this mixture contains all the water vapor it can hold, the air is termed saturated. The amount, or weight, of water vapor which can mix with air in a saturated condition depends upon the temperature. As the temperature increases, the weight of the moisture which can be held in suspension in the surrounding air-water vapor mixture also increases. There is, nevertheless, no direct relation between these two quantities but there is a direct ratio between vapor pressure and the weight of moisture in such mixtures. The temperature which we have been discussing is known as the dry bulb temperature because it is measured, usually, with a mercury thermometer that has a bulb that is dry on its outer surface.

When air-water vapor mixtures contain less moisture than they can hold, there is a definite relation established with the possible saturated condition which can be expressed in percentage, based on 100 per cent of saturation. This is known as relative humidity. There is much confusion between the relative and the absolute humidity. This confusion will not exist if it is remembered that the absolute humidity is the actual weight of moisture present in the air-water vapor mixture and is never expressed in percentage.

To measure the relative humidity by means of temperature alone it is necessary to obtain the dry bulb temperature at the same instant that the temperature of evaporation is read. This latter temperature is known as the wet bulb temperature.

There is a technique in obtaining correct wet bulb temperatures which will be explained in a later article on measurement methods. The difference between the wet and the dry bulb readings, or the wet bulb depression, may be used to obtain the relative humidity in the psychrometric tables of the U. S. Weather Bureau. Also, by following the dry and wet bulb temperatures to their intersection on psychrometric charts it is possible to read the relative humidity.

The two psychrometric charts presented in Figs. 2 and 3, will furnish the necessary illustration of the method of reading relative humidities from such charts. For example, take a dry bulb temperature of 95 and a wet bulb temperature of 58 deg. fahr. At the intersection of the two lines it is seen that the relative humidity is approximately 5 per cent. Directly to the left of the intersection, at the margin, the weight of moisture for this mixture is read as 13 grains per pound of dry air. Also, from the intersection of the dry bulb temperature with the 100 per cent relative

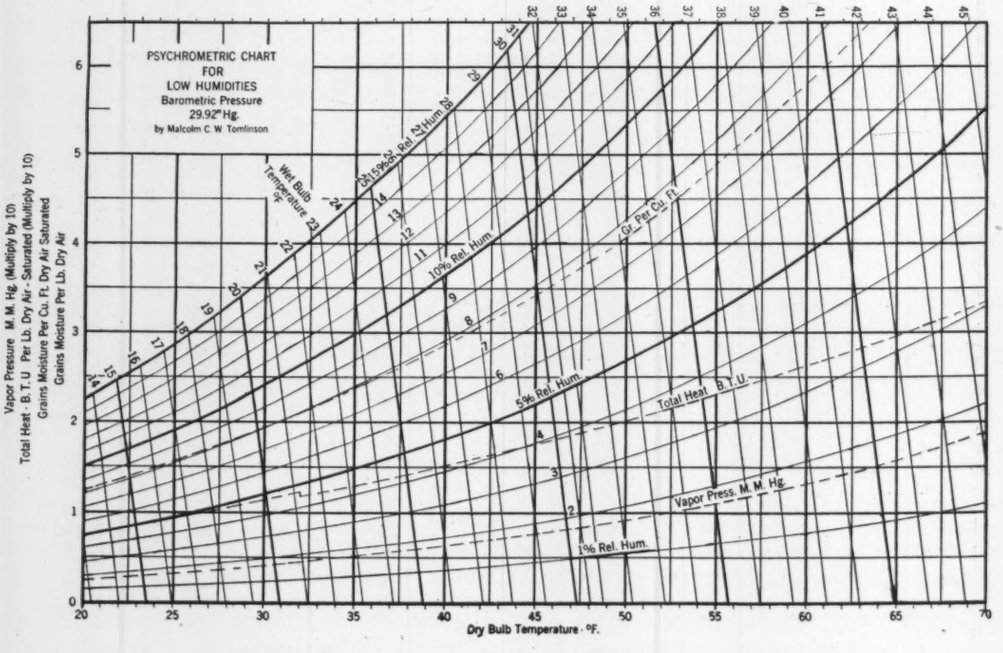

PSYCHROMETRIC CHART
FOR
LOW HUMIDITIES
Barometric Pressure
29.92" Hg.
by Malcolm C. W. Tomlinson
Vapor Pressure - M.M. Hg. (Multiply by 10)
Total Heat - B.T.U. Per Lb. Dry Air - Saturated (Multiply by 10)
Grains Moisture Per Cu. Ft. Dry Air Saturated
Grains Moisture Per Lb. Dry Air
Wet Bulb Temperature °F.
Gr. Per Cu. Ft.
10% Rel. Hum.
5% Rel. Hum.
Total Heat B.T.U.
Vapor Press. M.M. Hg.
1% Rel. Hum.
Dry Bulb Temperature - °F.

FIG. 2.—PSYCHROMETRIC CHART FOR LOW HUMIDITIES

humidity line in the psychrometric charts furnished in Fig. 4 and Fig. 5, the weight of the moisture in the saturated mixture, for the same dry bulb temperature, is read as approximately 258 grains per pound of dry air. It is easily seen, by dividing the weight of the saturated moisture into the weight of the moisture in the actual mixture that the relative humidity is practically 5 per cent.

The Subject of Low Humidities Concerns Every Human Being

It is now possible to take up the subject of human comfort, with special attention to low humidities, and this will be found a much more interesting subject since it vitally concerns every human being.

For many years the medical profession has sent patients with tuberculosis to dry climates such as Arizona. Nevertheless, the first proposal to air condition a large factory room at 10 per cent relative humidity, a dry bulb temperature of 85 deg. fahr. and air motion of 100 feet a minute was received with expressions of doubt, if not disapproval, by the doctors, even though the employees were to be in the room only a few hours a day. The main objection raised was that the differences in temperature and pressure between the conditioned room and the surrounding factory might affect the heart. Therefore, only young men free from heart diseases were employed for such purposes. Naturally, absences due to sickness were carefully noted. It soon became evident that these employes, although working without coats, vests and shirts, had no tendencies towards colds. As the conditioned room was most comfortable in warm and cold weather other employees tended to visit it. To eliminate this practice the temperature was gradually raised to 115 deg. fahr. Later, when a new room was conditioned down to ½ per cent relative humidity at the same air motion, the temperatures were again raised from 85 to 95 deg. fahr. because, at the lower temperature, the room was cool to persons normally clothed. It was then raised from 95 to 115 deg. fahr. for the same reason as with the 10 per cent room.

About the time the first development work was under way in the application of low humidities to actual factory conditions the research laboratory of the American Society of Heating and Ventilating Engineers, in conjunction with the Bureau of Mines and the U. S. Public Health Service, began to study effect of temperature, relative humidity and air motion on human comfort. The earlier investigations proved rather conclusively that the pulse rate determines the extent of human comfort or discomfort, that sweat was the chief factor in producing mental irritation, that ice water did not produce cramps under conditions of high temperatures and humidities, that the systolic and diastolic blood pressure fell when the temperature and humidity were increased moderately and that the systolic pressure rose, while the diastolic fell, in high temperatures and humidities. Under the latter conditions human subjects complained of headaches, became irritable and soon developed palpitation of the heart. This work indicated that human beings could stand a temperature of 112 deg. fahr. at 100 per cent relative humidity for a half hour and that this condition was, approximately, as unendurable as 157 deg. fahr. at 15 per cent for the same period. It thus became apparent that the physiological reactions of human beings to the temperatures and humidities which were being used in the two conditioned rooms were not adverse to the maintenance of healthful working conditions.

Effective Temperature

The next step of the research laboratory of the A. S. H. & V. E. was to correlate the vast amount of comfort data gathered. This was first done on a psychrometric chart of the Carrier type. Then, since a common reference factor was needed to enable one to make comfort comparisons at various temperatures, relative

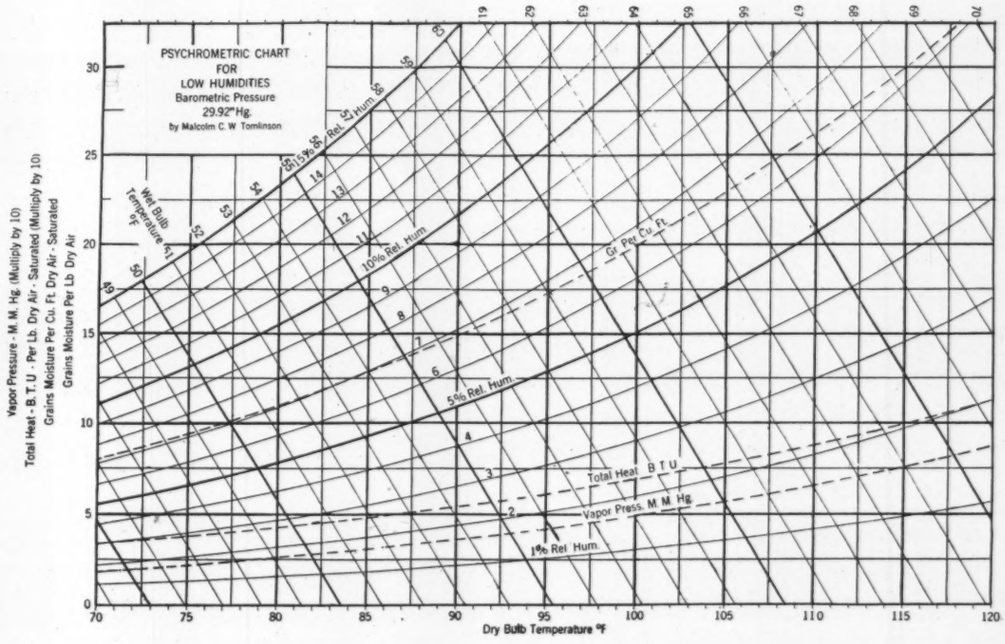

PSYCHROMETRIC CHART
FOR
LOW HUMIDITIES
Barometric Pressure
29.92" Hg.
by Malcolm C. W. Tomlinson
Vapor Pressure - M. M. Hg. (Multiply by 10)
Total Heat - B. T. U. - Per Lb. Dry Air - Saturated (Multiply by 10)
Grains Moisture Per Cu. Ft. Dry Air - Saturated
Grains Moisture Per Lb. Dry Air
Wet Bulb Temperature °F
Gr. Per Cu. Ft.
10% Rel. Hum.
5% Rel. Hum.
1% Rel. Hum.
Total Heat B T U
Vapor Press. M M Hg.
Dry Bulb Temperature °F

FIG. 3.—PSYCHROMETRIC CHART FOR LOW HUMIDITIES

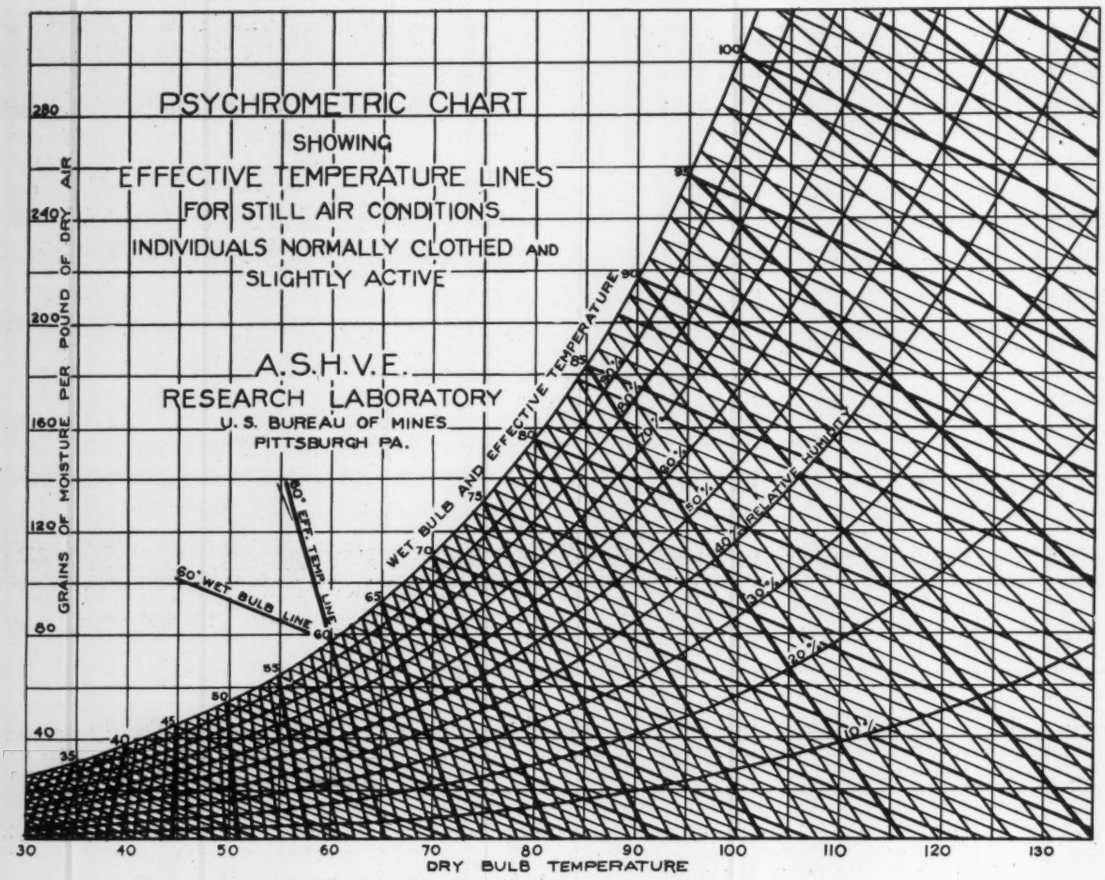


FIG. 4.—A. S. H. & V. E. PSYCHROMETRIC CHART

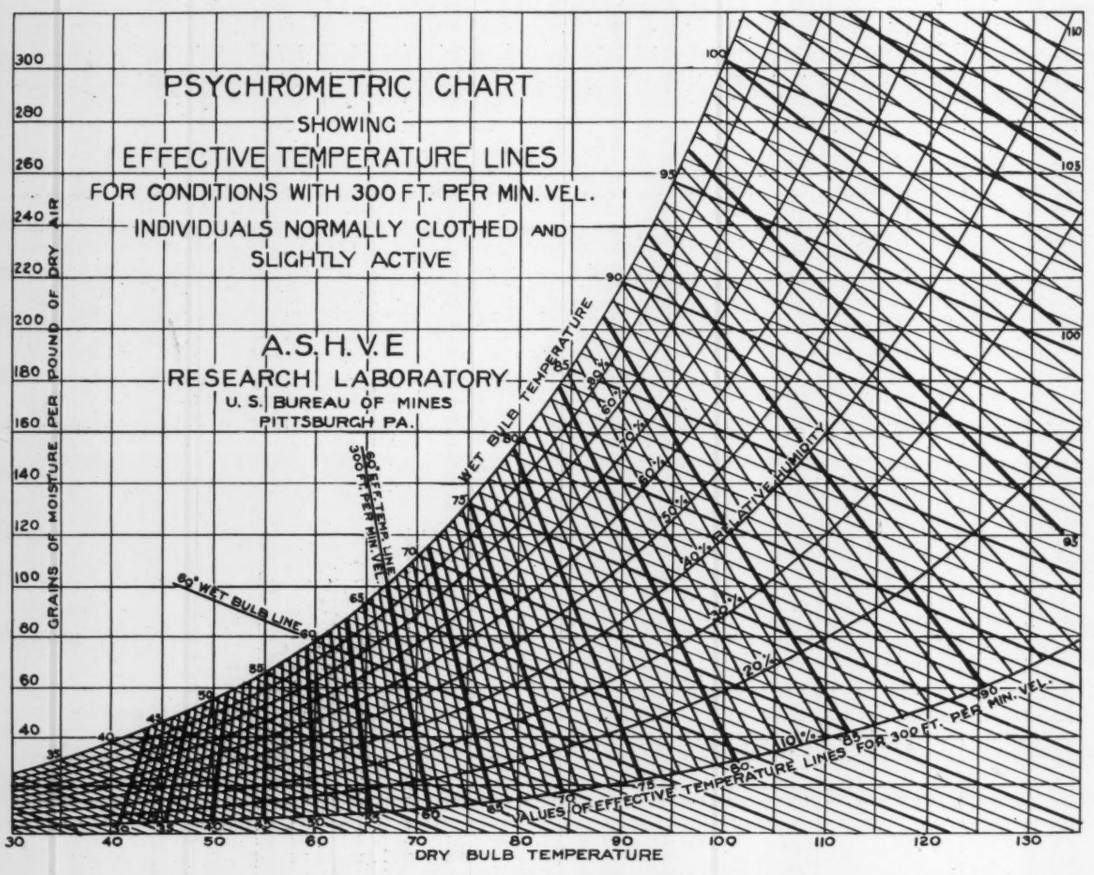

PSYCHROMETRIC CHART
SHOWING
EFFECTIVE TEMPERATURE LINES
FOR CONDITIONS WITH 300 FT. PER MIN. VEL.
INDIVIDUALS NORMALLY CLOTHED AND
SLIGHTLY ACTIVE
A.S.H.V.E
RESEARCH LABORATORY
U. S. BUREAU OF MINES
PITTSBURGH PA.
60° EFF. TEMP. LINE 300 FT. PER MIN. VEL.
60° WET BULB LINE
WET BULB TEMPERATURE
RELATIVE HUMIDITY
VALUES OF EFFECTIVE TEMPERATURE LINES FOR 300 FT. PER MIN. VEL.
GRAINS OF MOISTURE PER POUND OF DRY AIR
DRY BULB TEMPERATURE

FIG. 5.—A. S. H. & V. E. PSYCHROMETRIC CHART

TABLE 1—HUMAN COMFORT CONDITIONS FOR INDIVIDUALS NORMALLY CLOTHED SLIGHTLY ACTIVE

REL. HUM.—%	LOWER LIMIT FOR EACH COMFORT ZONE					
	0	20	40	60	80	100
	DRY BULB TEMPERATURES					
Still Air Condition						
Best Zone	65	64	63	62	61	60
Good Zone	78	76	73	71	69	68
Fair Zone	91	87	83	80	77	75
Poor Zone	107	99	93	88	85	82
100 F.P.M. Condition						
Best Zone	68	67	66	65	64.5	63.5
Good Zone	80	78	75.5	73.5	71.5	70.5
Fair Zone	93	89	85	82.5	79	77
Poor Zone	108.5	100.5	94.5	89.5	86.5	83.5
200 F.P.M. Condition						
Best Zone	69.5	68.5	67.5	66.5	66	65.5
Good Zone	81	79	77	75	73	72
Fair Zone	93.5	89.5	87	83.5	80	78
Poor Zone	109.5	101.5	95.5	91	87.5	84.5
300 F.P.M. Condition						
Best Zone	71	70	69	68	67.5	67
Good Zone	82	80	78	76	75	73
Fair Zone	94	90	86.5	84	81	79
Poor Zone	110	102	96	92	88	85
500 F.P.M. Condition						
Best Zone	73.5	72	71.5	70.5	70	69
Good Zone	83.5	81.5	80	78.5	76.5	75
Fair Zone	94.5	90.5	87	85.5	82.5	80.5
Poor Zone	111	103	97.5	93.5	89	86

Note: The upper limit for each comfort zone corresponds to the lower limit of the zone which is next poorer in desirability.

The usual air motion is 50 F. P. M. and, when a wind is stirring outside (windows open), the air may be moving at 100 F. P. M. inside. Therefore, still air conditions are not often found.

humidities and rates of air motion, a new term—"effective temperature"—was devised. Effective temperature may be defined as the scale which is a true index of a person's feeling of warmth of coolness in all combinations of temperature, air motion and humidity. These effective temperature lines were plotted, or superimposed, on the psychrometric chart already prepared.

Two of the A. S. H. & V. E. psychrometric charts are shown in Fig. 4 and Fig. 5. Both are for individuals normally clad. The first applies to the condition of still air and the second to air at 300 feet per minute. Also, in Table 1, the dry bulb temperatures for best, good, fair and poor human comfort at various humidities and air motions are given for those normally clad. It is possible to secure corresponding data on individuals stripped to the waist, on the amount of work which can be performed by human beings at different effective temperatures and on the variation of human comfort at the two extreme seasons of the year. This information can be had through the A. S. H. & V. E. Those who are interested should avail themselves of the opportunity to study the literature on this subject.

All of these investigations have covered relative humidities between 20 and 100 per cent. It is, therefore, of interest to note the actual factory experiences under lower humidity conditions. With temperatures of the dry bulb between 85 and 115, air motion of 100 feet per minute and ½ per cent relative humidity, the effective temperatures are at least 8 degrees lower than those plotted on the A. S. H. & V. E. charts. For the same conditions, but at 10 per cent relative humidity, the effective temperatures are still slightly lower than those charted.

In order that no misapprehensions will be entertained in regard to the present attitude of the medical authorities toward low humidities it is interesting to note that experiments on the use of conditioned air are under way in two hospitals at least. In one the application is being made to pneumonia. Here the original experiments started at 40 per cent relative humidity and normal room temperatures. At present they are working at 30 per cent simply because it is impossible, with their present equipment, to go lower. It is the belief of the author that it will not be long before many hospitals and sanitariums will be treating all forms of lung troubles with air conditioning at humidities not higher than 10 per cent

Shore to Ship Steam Service

By E. W. Sylvester, Lt., Construction Corps, U. S. N.

IT IS often either necessary or desirable to supply steam from a shore plant to a ship for heating, galley service, and the operation of steam driven auxiliaries when the vessel is undergoing overhaul involving temporary decommissioning of her own steam plant. Similar shore service is required for heating vessels building and vessels under overhaul which are not normally fitted with steam or other adequate heating systems.

Steam is supplied from ashore to vessels at ship yard piers or in drydocks from the central power plant or the central heating plant of the yard, or from steam plants built for or adapted to the purpose. In most ship yards steam mains are run from the central power plant to each pier and dock for supplying this temporary service when required. In some yards involved in extensive construction programs, such as the building of a vessel like the airplane carrier Lexington, the magnitude of the heating problem necessitates the construction and operation of a steam plant designed expressly for heating vessels during their building periods. Some establishments have found it expedient, in cases requiring only a moderate heating capacity, to provide temporary steam service to ships by the use of donkey boilers.

Shore to ship steam connections must necessarily be flexible if the vessel is afloat to allow for vertical movement of the ship due to the rise and fall of the tide and for horizontal movements of the vessel within the limits determined by her mooring lines due to wind and current. This connection is usually made with a flexible

metallic steam hose or wire-inserted rubber steam hose. The rubber hose is, of course, preferable from the point of view of minimizing condensation losses in this necessarily exposed connection but the flexible metallic hose is extensively employed due to its durability.

A reducing valve and a stop valve are installed at the shore end of the shore to ship connection. The ship end is usually connected to temporary rigid piping aboard the vessel, which temporary piping is cut into the ship's auxiliary steam system through a spare connection, if available, or by breaking a joint in the ship's system for the purpose. Some vessels are fitted with permanently installed deck connections from their auxiliary steam systems for receiving dock steam service.

The shore reducing valve on temporary steam connections is usually set to supply steam at the maximum pressure which will be required on the vessel, and permanently installed reducing valves in the branches to the various subsystems control the pressures to these subsystems. The systems for which steam is usually supplied to a vessel under overhaul are steam heating, hot water, galley pressure cookers and urns, scullery dish-washing machinery, steam laundry machinery and, occasionally, steam winches. In connection with this latter service, it happened at one yard that the services of the submarine lifting machinery of a tender, which was at the yard under overhaul and whose fires had been hauled, where desired for lifting the bow of a submarine for inspection of the torpedo tubes. Arrangements were therefore made to supply steam to the tender's lifting machinery from the central power plant of the yard.

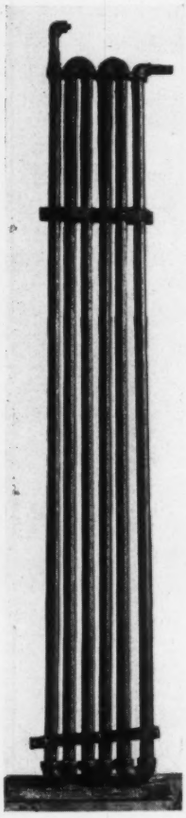


FIG. 2.—A REMOVABLE RADIATOR USED FOR HEATING SUBMARINES WHILE IN DRYDOCK AT THE PORTSMOUTH NAVY YARD

FIG. 1.—TEMPORARY STEAM SERVICE TO SHIPS IN DRYDOCK AT THE CHARLESTOWN NAVY YARD IS SUPPLIED BY THIS PORTABLE DONKEY BOILER, WHICH WILL WITHSTAND 100 LB. PRESSURE

Arrangement at Portsmouth Navy Yard

An unusual instance of shore to ship steam service is afforded by the arrangements utilized for heating submarines under overhaul at the Portsmouth Navy Yard during the winter months. Only the largest and latest of our submarines are fitted with permanently installed steam heating systems. The great majority are provided with portable electric heaters only, which are inadequate to provide a satisfactory degree of comfort to crew and yard workmen during severe weather. The yard has developed what is in effect a portable steam heating system for this service. The radiators are made up of six parallel ten foot lengths of 1¼-in. iron pipes connected in series by 180 deg. fittings. These made-up radiator sections can be readily shipped through a submarine's deck hatches, and they are installed in the habitable compartments of the vessel by suspending them from convenient points of attachment. A temporary steam supply line is run the length of the vessel through the bulkhead doors to supply these radiators. This supply line is connected to the dock steam main through the usual flexible metallic steam hose. The radiators are connected in parallel when it is practicable to trap them. Under conditions of especially severe weather or when the physical conditions aboard the submarine render the installation of traps impracticable, several radiators are connected in series and permitted to exhaust to the atmosphere, the reducing valve on the dock being set to a pressure which will just keep a feather of steam blowing through the exhaust.

Another unusual instance of shore to ship steam service arrangements has been reported to the writer in connection with certain German wartime submarines which are reported to have been fitted with permanently installed steam heating systems solely for the purpose of heating these vessels from the dock when they are in port.

A veteran of the service is the donkey boiler which is shown in Fig. 1. When it is necessary to supply steam to a vessel undergoing repairs at the Charlestown Navy Yard, this boiler is called to the colors.

Engineering

The following definition of "engineering," designed to be general enough to include the various branches, has been proposed by Col. P. E. Barbour, secretary of the Mining and Metallurgical Society of America: "Engineering is the judicious application of the technical sciences to the human solution of inanimate mechanical problems."

Heating·Piping
and Air Conditioning

American Society of Heating and Ventilating Engineers JOURNAL SECTION

AUGUST 1929

Material in this Section prepared by Publication Committee: A. C. Willard, *Chairman*; N. W. Downes, and W. A. Rowe. : A. V. Hutchinson, Manager of Publications

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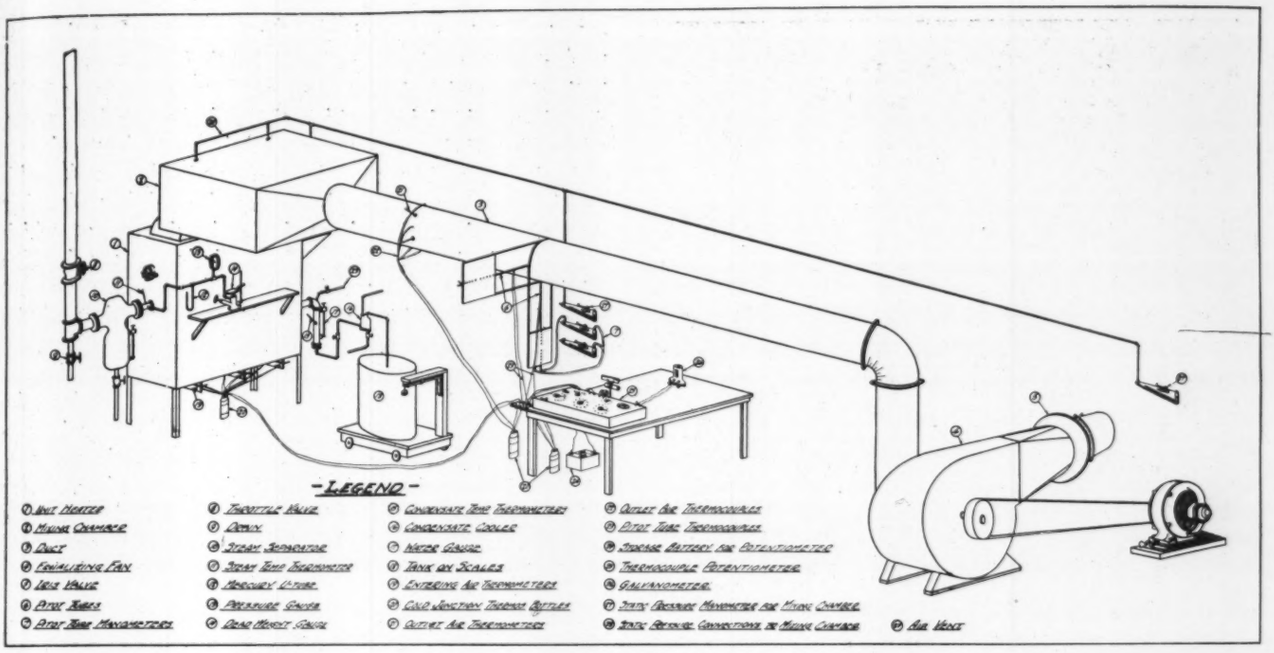


FIG. 1.—DIAGRAM OF A FLOOR TYPE UNIT HEATER SHOWING TEST EQUIPMENT

Heat and Air Volume Output of Unit Heaters

The results of co-operative research work between the A. S. H. V. E. and the University of Kentucky

By L. S. O'Bannon[1], Lexington, Ky.
MEMBER

THIS paper describes a series of sixteen tests made on three unit heaters for the purpose of supplying information to the AMERICAN SOCIETY OF HEATING and VENTILATING ENGINEERS' Committee on Code for Testing Unit Heaters. The work is the result of a cooperative agreement between the University of Kentucky and the Research Laboratory of the Society.

The tests were supervised by the author of this paper in co-operation with his assistants and the Director of the Research Laboratory. A full measure of credit is due the men who constituted the crew of testers and who performed well all the duties of their office. These men were R. W. Bozeman, J. C. Lindley, L. L. Massie, J. W. May and E. D. Moore, all of whom were members of the 1929 graduating class of the University of Kentucky.

In testing unit heaters the information usually desired is heat output, air volume output and final air temperature for several fan speeds and for various degrees of entering air temperature.

There is only one generally approved method of measuring heat output. This method consists of operating the unit for a short period of time, usually about one hour, and measuring the quantity of steam condensed; the pounds of steam condensed per hour multiplied by the heat given up per pound of steam gives the heat output of the unit in B. t. u. per hour.

There are two methods in common use for measuring the air volume output. One consists of measuring the air directly by means of some type of flow meter, usually the Pitot tube. The other is called the condensate-temperature rise method. The latter is based on the assumption that the heat absorbed by the air is equal to the heat given up by the steam. The fundamental formula for this method is

$$W_c\ (H-q)=W_a C\ (T_F-T_E)$$

Where W_c=weight of condensed steam in pounds per hour

H=total heat content of steam entering the heater, B.t.u. per pound

q=heat in the condensate leaving the heater, B.t.u. per pound

[1]Professor of Heat Engineering, University of Kentucky, Lexington, Ky. Presented at the Semi-Annual Meeting o the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Lake-of-Bays, Ontario, Canada, June, 1929.

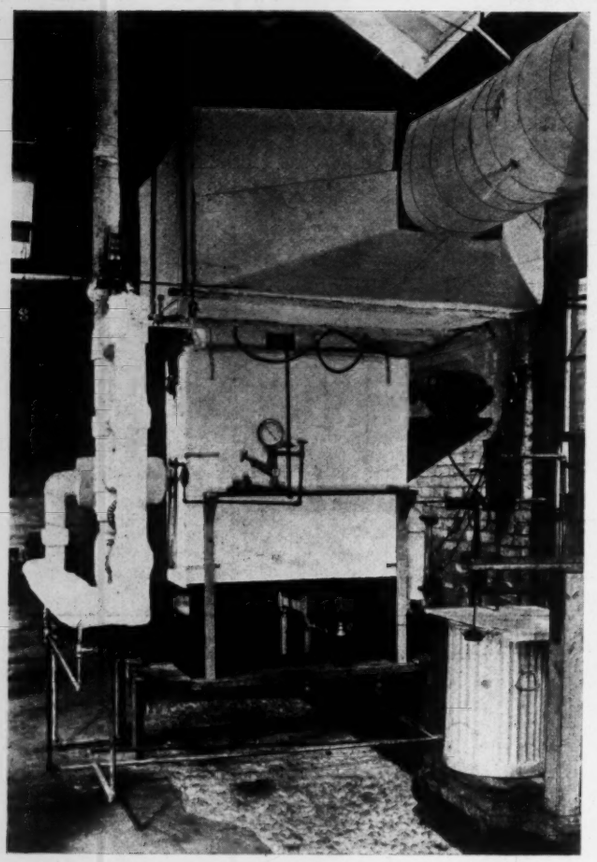


FIG. 2.—VIEW OF UNIT HEATER AND MIXING CHAMBER

W_a=weight of air passing through the heater in pounds per hour.

C=specific heat of air in B.t.u. per lb. per deg. fahr. (usually assumed as approximately 0.24)

T_F=final temperature of the air leaving the heater, deg. fahr.

T_E=temperature of air entering the heater, deg. fahr.

The volume of the air for any unit of time and for any specified pressure and temperature is calculated from the weight of the air, W_a, which is obtained from the formula after substituting numerically the other values derived from the tests.

W_c is obtained directly by weighing the condensed steam. By referring to steam tables, H is obtained from the steam temperature and pressure observed during the test, and likewise, q is obtained from the temperature of the condensate. These values are easily and accurately obtained by ordinary methods. T_E is the average of the readings of several thermometers or thermocouples arranged around the inlet to the heater. As a rule no difficulty is encountered in measuring the entering air temperature. The principal precaution required is to shield the thermometers from the direct radiation from the heater. T_F is more difficult to determine accurately due to the non-uniform temperature and velocity of the warm air usually encountered at the outlet or outlets of the unit. To eliminate temperature indications which do not represent a true average a large duct is provided into which the unit heater discharges. The enlarged duct serves as a mixing chamber and as a collecting chamber also in the case of a unit with multiple outlets. The air leaves the mixing chamber through a single opening of restricted area at which point the final temperature of the air is measured. The mixing chamber and unit heater are insulated to prevent excessive loss of heat and consequent lowering of the final air temperature.

To overcome the resistance of the mixing chamber an auxiliary fan is required. The fan is connected to the outlet of the mixing chamber and the air flow is regulated by means of a damper so that a static pressure of zero is maintained within the mixing chamber. With this arrangement it is supposed that the unit is operating under the same conditions of free delivery as would obtain if no duct work were attached to the unit heater outlet.

With the Pitot tube method of finding air volumes it is still desirable to know the final air temperature. The same scheme of discharging into a mixing chamber is used and the Pitot tube measurements are made in the duct between the mixing chamber and the equalizing fan.

Fig. 1 shows a diagram of a floor type of unit heater with test equipment arranged to conform to the methods of testing just outlined. The diagram illustrates the set-up used in this investigation. Several photographic views are shown in Fig. 2-5.

In a code for testing it is desirable to have a single method of procedure. Where two methods are equally in favor the question naturally arises as to which is the more accurate. Not having a third method of absolute accuracy it is impossible to conclude directly from experiment as to the accuracy of the two methods in question. However, the two methods may be compared one with the other; advantages and disadvantages of each weighed, and the probable errors evaluated.

This investigation was undertaken from the latter standpoint. Nothing unusual or ultra-precise was attempted. Reasonable effort was made to eliminate sources

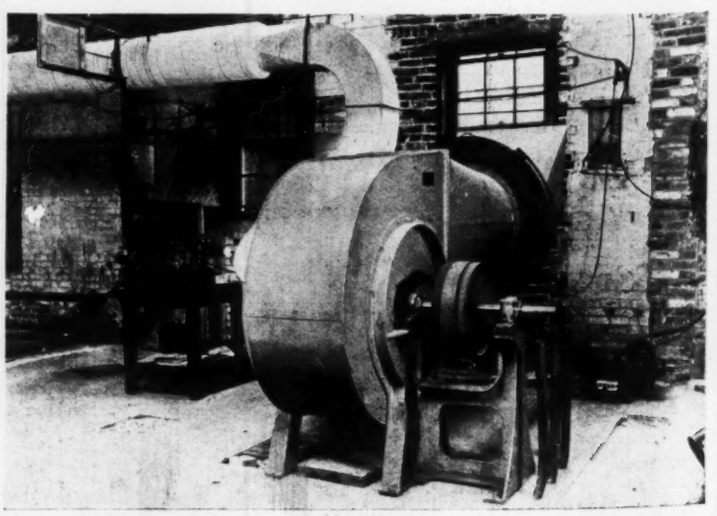


FIG. 3.—VIEW SHOWING PITOT TUBE ASSEMBLY, POTENTIOMETER EQUIPMENT AND EQUALIZING FAN

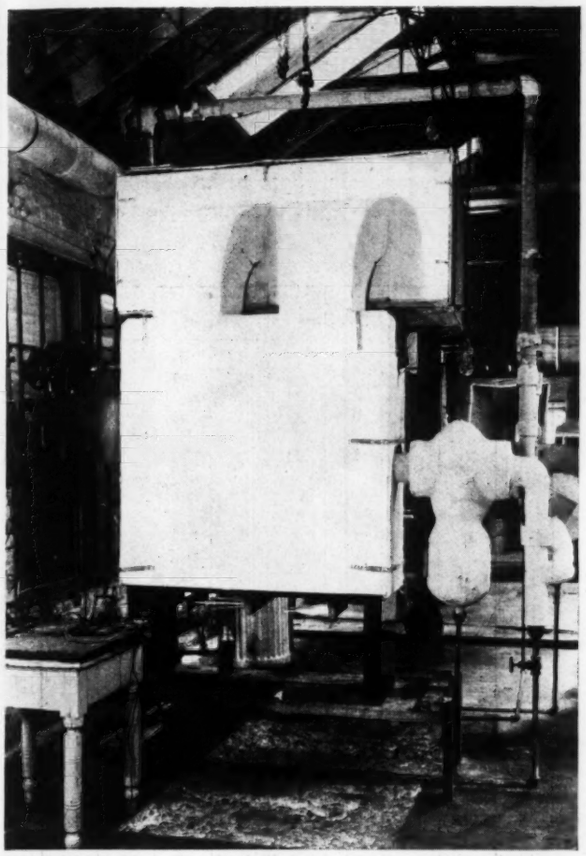


FIG. 4.—REAR VIEW OF UNIT HEATER AND MIXING CHAMBER

of error and a few obvious corrections were estimated and applied. The arrangement of the set-up was planned and the application of instruments was executed with no greater elaboration of detail than ordinarily would have been observed in any test of similar nature.

The results of the tests have been summarized in Table 1. It will be necessary to explain some of the items of this summary in detail, following the order in which they are listed, and keeping in mind the chief purpose of the investigation, that is, a comparison of the Pitot tube method with the condensate-temperature rise method of measuring air volume output. Tests numbered 1 to 6 inclusive were all made on the same unit heater which was of the draw-through type with centrifugal fan. Test numbers 11 to 16 inclusive were made on another unit of the same type but of different manufacture. Each of these units had two outlets. Tests numbers 7 to 10 inclusive were made on a blow-through heater with disc fan and a single outlet.

The duration of each test was more or less arbitrary. When five or six readings at 10 minute intervals gave repeatedly practically the same values nothing was to be gained by continuing the tests for a longer period.

A constant steam pressure was maintained to give a saturation temperature of 227 deg. fahr. which corresponds to 5 lb. gauge pressure at standard barometric pressure. The steam was supplied from a high pressure boiler and was reduced to the required pressure through a throttle valve operated manually. The steam was maintained at a few degrees superheat to eliminate possible error due to moisture in the steam. During some of the tests it was necessary to inject a slight spray of water into the steam to limit the degree of superheat.

At the outlet of the heater the drawing, Fig. 1, shows a vertical pipe with a *Y* fitting and a water gauge. The condensate thermometer was placed in a well screwed into the *Y*. The level of the condensate was held a few inches above the thermometer bulb. A stem correction was applied to this thermometer. To insure elimination of air a vent valve was attached to the outlet of the heater and a thread of steam was allowed to flow through the valve continually. The heat given up per pound of steam was calculated as the difference between the heat content of the steam at inlet conditions minus the heat of the liquid at the temperature indicated by the condensate thermometer.

The condensate was weighed on platform scales capable of being read to ¼ of a pound.

Both thermometers and thermocouples were used to measure entering and final air temperatures. Six thermometers, and six thermocouples connected in parallel, were arranged around the inlet to the heater. Four thermometers, and four thermocouples in parallel, were used at the outlet of the mixing chamber. No correction of any kind was applied to the inlet temperature readings. The inlet thermometers and thermocouples were shielded from the heater but they were not shielded from surfaces in the room (the floor for instance) which may have been at a lower temperature than the entering air.

The final air temperatures, by thermometers, recorded

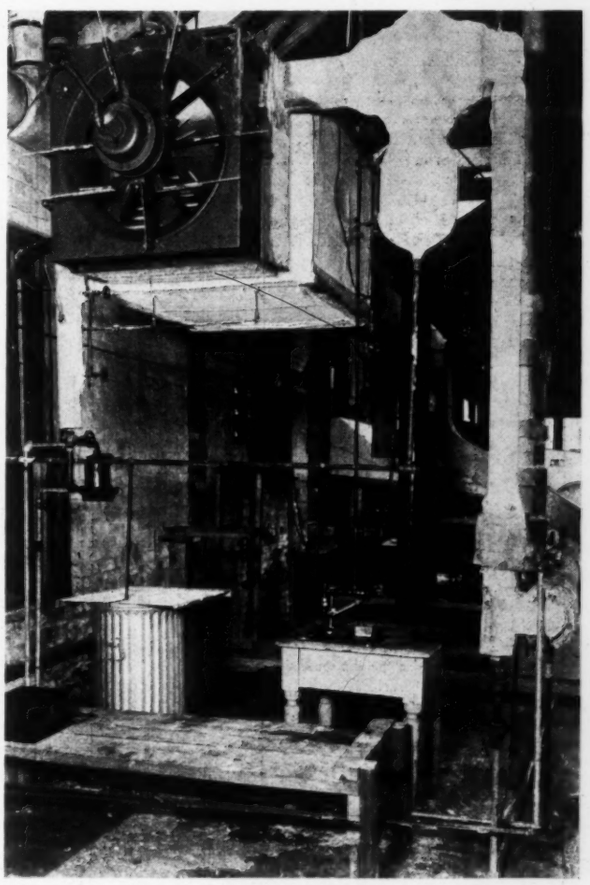


FIG. 5.—VIEW SHOWING CEILING TYPE OF UNIT MOUNTED FOR TEST

in the summary include a correction for emergent stem and also a positive correction for heat loss by conduction through the insulation on the mixing chamber and the unit heater casing. The emergent stem correction in every test was approximately 0.2 deg. fahr. and was positive in sign. Variations from this value for individual tests were in hundredths of a degree only. The correction for heat losses amounted to from 0.5 deg. fahr. to 2.5 deg. fahr. depending upon the final temperature of the air. The correction was estimated by merely calculating the transmission loss through the insulation using the temperature rise of the air as the temperature difference and neglecting the influence of air velocity on the inside of the heater and mixing chamber. This correction was also applied to the final air temperatures obtained from the thermocouple readings.

TABLE 1.—SUMMARY OF TESTS ON UNIT HEATER

1	Test Number	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
2	Duration of Test, Minutes	50	40	50	50	50	50	40	40	40	40	50	40	40	40	40	40
3	Fan Speed, Rev. per Minute	1788	1181	792	1778	1165	797	1750	1750	1750	1750	834	436	647	841	647	437
4	Steam Pressure, Lb. per Sq. In	5.4	5.4	5.4	5.6	5.6	5.6	5.5	5.5	5.5	5.5	5.4	5.4	5.4	5.4	5.4	5.4
5	Steam Temperature, Deg. F	233	230	232	234	236	231	233	236	236	246	238	232	238	232	230	230
6	Heat Content of Steam, B.T.U. per Lb	1160.3	1158.3	1159.8	1160.8	1161.8	1159.5	1160.3	1161.8	1161.8	1166.6	1162.7	1159.8	1162.7	1159.8	1158.8	1158.8
7	Condensate Temperature, Deg. F	225	225	225	224	224	224	224	224	224	224	224	225	225	225	225	226
8	Heat of Liquid at Condensate Temperature, B.T.U.	193.1	193.1	193.1	192.1	192.1	192.1	192.1	192.1	192.1	192.1	192.1	193.1	193.1	193.1	193.1	194.1
9	Heat Given up by Steam, B.T.U. per Lb	967.2	965.7	966.7	968.7	969.7	967.4	968.2	969.7	969.7	974.5	970.6	966.7	969.6	966.7	965.7	964.7
10	Weight of Condensate, Lb. per Hour	204.3	146.6	105.6	212.1	150.3	111.6	135.8	135.0	97.5	157.1	192.9	116.6	159.4	198.4	162.8	120.0
	Temperature of Entering Air Deg. F.—																
11	Thermometer No. 1	78.8	81.3	83.5	70.7	73.3	73.3	76.1	76.1	77.7	75.3	70.1	71.2	70.4	69.2	69.0	69.7
12	Thermometer No. 2	78.9	81.8	84.2	71.4	73.5	74.1	77.2	77.0	78.6	76.5	70.3	71.5	70.0	68.6	68.5	69.0
13	Thermometer No. 3	78.1	80.5	82.5	71.0	72.9	72.8	83.6	83.2	83.6	81.4	70.2	71.5	70.1	69.2	68.9	69.7
14	Thermometer No. 4	78.1	80.5	83.0	70.1	72.4	72.9	82.7	82.0	83.2	80.3	70.9	71.8	70.7	69.9	79.7	69.6
15	Thermometer No. 5	78.0	80.5	82.6	70.5	73.0	73.3	78.1	58.4	81.0	77.7	70.9	71.7	70.5	69.6	69.1	70.0
16	Thermometer No. 6	77.7	80.5	82.6	70.4	72.6	72.7	76.7	76.8	79.1	75.8	71.3	71.7	70.5	69.8	69.5	69.2
	Average Temperature of Entering Air—																
17	Thermometers	78.3	80.8	83.1	70.7	73.0	73.2	79.1	78.8	80.5	77.8	70.6	71.6	70.4	69.4	69.1	69.5
18	Thermocouples	78.8	81.6	84.2	71.5	73.7	74.1	79.2	79.6	80.7	78.5	71.1	72.2	70.9	70.1	69.6	70.3
	Final Temperature of Air, Deg. F—(including Stem Correction and Temperature Drop Due to Heat Losses)—																
19	Thermometer No. 1	139.9	149.3	157.1	135.6	143.7	151.5	146.7	144.8	144.5	140.7	147.4	162.7	153.5	147.3	153.0	162.5
20	Thermometer No. 2	140.4	149.9	157.8	136.2	144.4	151.9	144.6	147.0	150.5	142.1	148.1	163.8	154.4	148.0	153.9	163.5
21	Thermometer No. 3	138.9	148.0	156.0	134.6	142.6	149.9	145.4	145.9	146.5	141.3	147.8	163.5	154.0	147.9	153.8	162.5
22	Thermometer No. 4	139.1	148.2	155.5	134.9	142.7	150.2	145.9	146.5	153.7	139.4	147.7	163.6	153.9	147.9	153.8	162.9
	Average Final Temperature of Air—																
23	Thermometers	139.6	148.9	156.6	135.3	143.4	150.9	145.7	146.1	148.8	140.9	147.8	163.4	154.0	147.8	153.6	162.9
24	Thermocouples	140.1	148.8	156.9	135.6	143.9	151.4	144.7	145.2	149.1	140.3	147.1	162.8	153.0	146.8	154.0	162.5
	Temperature Rise, Deg. F.—																
25	By Thermometers	61.3	68.1	73.5	64.6	70.4	77.7	66.6	67.2	68.3	63.1	77.2	91.8	83.6	78.4	84.5	93.4
26	By Thermocouples	61.3	67.2	72.7	64.1	70.2	77.3	65.5	65.6	68.4	61.8	76.0	90.6	82.1	76.7	84.4	92.2
27	Psychrometer Reading—Dry Bulb							77.5	77.5	77.5	77.0	72.0	72.0	71.0	71.0	71.0	71.0
28	—Wet Bulb							66.0	66.0	66.0	66.5	58.5	58.5	58.5	58.5	58.0	58.0
29	—Relative Humidity, %							55.0	55.0	55.0	58.0	44.0	44.0	47.0	47.0	45.0	45.0
30	Mean Specific Heat including Moisture, B.T.U./Lb./Deg. F	0.2421	0.2422	0.2422	0.2420	0.2421	0.2421	0.2445	0.2445	0.2446	0.2446	0.2437	0.2438	0.2438	0.2438	0.2437	0.2437
	Weight of Air, Lb. per Hour—																
31	By Thermometer	13314.	8583.	5734.	13143.	8551.	5739.	8074.	7968.	5659.	9919.	9952.	5036.	7853.	10034.	7634.	5085.
32	By Thermocouples	13314.	8698.	5798.	13245.	8573.	5769.	8210.	8162.	5651.	10127.	10109.	5103.	7722.	10257.	7644.	5152.
	Average of Pitot Tube Readings—																
33	Static Pressure, in H_2O	−0.711	−0.297	−0.130	−0.698	−0.298	−0.133	−0.31	−0.312	−0.053	−0.580	−0.916	−0.240	−0.539	−0.924	−0.539	−0.290
34	Velocity Pressure, in. H_2O	0.643	0.267	0.112	0.638	0.263	0.113	0.236	0.233	0.113	0.368	0.371	0.093	0.209	0.378	0.217	0.094
35	Barometer Reading, Inches of Mercury	29.15	29.15	29.15	28.82	28.82	28.82	28.88	28.88	28.87	28.87	29.07	29.09	29.09	29.10	29.11	29.12
36	Absolute Pressure of Air at Pitot Tube, in. Hg	29.10	29.13	29.14	28.77	28.80	28.81	28.86	28.86	28.87	28.83	29.00	29.07	29.05	29.03	29.07	29.10
37	Temperature of Air at Pitot Tube, Deg. F., by Thermocouples	138.2	147.0	153.7	134.4	142.0	147.6	143.0	143.1	146.5	138.7	145.9	159.3	151.2	145.9	151.0	159.0
38	Density of Air at Pitot Tube, Lb. per Cu. Ft.	.06457	.06370	.06302	.06424	.06350	.06293	.06310	.06309	.06276	.06346	.06325	.06203	.06279	.06288	.06286	.06211
39	Velocity of Air at Pitot Tube, Ft. per Min.	3459.	2244.	1461.	3455.	2231.	1469.	2120.	2107.	1471.	2640.	2655.	1342.	2000.	2688.	2037.	1349.
40	Area of Duct at Pitot Tube, Sq. Ft.	1.01714	1.01714	1.01714	1.01714	1.01714	1.01714	1.01714	1.01714	1.01714	1.01714	1.01714	1.01714	1.01714	1.01714	1.01714	1.01714
41	Cubic Feet of Air per Min. at Pitot Tube, C.F.M.	3518.	2282.	1486.	3514.	2269.	1494.	2156.	2143.	1496.	2685.	2701.	1365.	2034.	2734.	2072.	1372.
42	Static Pressure in Mixing Chamber, in. H_2O	0	0	0	0	0	0	0	0	+0.1	−0.1	0	0	0	0	0	0
43																	
44	Total Heat Output, B.T.U. per Hour	197599.	141572.	102084.	205461.	145746.	107962.	131482.	130910.	94546.	153094.	187229.	112717.	154554.	191793.	157216.	115764.
45	C.F.M. at 70 Deg. F. and 29.92 in. Hg., by Pitot Tube	3031.	1940.	1250.	3012.	1923.	1255.	1815.	1804.	1253.	2274.	2280.	1130.	1704.	2294.	1738.	1137.
	Ditto: Condensate—Temp. Rise Method—																
46	By Thermometers	2961.	1909.	1275.	2923.	1902.	1276.	1796.	1772.	1259.	2206.	2213.	1120.	1687.	2232.	1698.	1131.
47	By Thermocouples	2961.	1935.	1290.	2946.	1907.	1283.	1826.	1815.	1257.	2252.	2248.	1135.	1717.	2281.	1700.	1146.
	Percentage Condensate Method is High—																
48	Using Thermometer			2.00			1.67			0.48							
49	Using Thermocouples			3.20			2.23	0.61	0.61	0.32			0.44	0.76			0.79
	Percentage Condensate Method is Low—																
50	Using Thermometer	2.31	1.60		2.96	1.09		1.05	2.38		2.99	2.94	0.88	1.00	2.70	2.30	0.53
51	Using Thermocouples	2.31	0.26		2.19	0.83					0.97	1.40			0.57	2.18	
52																	
53																	
54																	
55																	

The mean specific heat of the air was calculated from the following formulae

For dry air,

$$C_p = 0.24112 + 0.000009\,t$$

For water vapor,

$$C_p = 0.4423 + 0.00018\,t$$

The specific heat values obtained from a combination of these formulae take into consideration not only the moisture content of the air but also the variation of specific heat with temperature. No psychrometer readings were taken during the first six tests and the air was assumed to be dry.

The weight of the air, items 31 and 32, was calculated from the fundamental formula for the condensate-temperature rise method already explained.

The Pitot tube was used according to standard practice. The duct was divided into five areas requiring 20 readings to complete a traverse. Only one traverse was made during a test. Two Pitot tubes were used, one for the horizontal traverse and one for the vertical traverse. Inclined tube draft gauges were used for measuring pressures. The gauges were checked for accuracy by comparison with a sensitive manometer patterned in principle after a micromanometer described by John L. Hodgson in the April 1929 issue of *Instruments*. The manometer is shown in Fig. 6. The design of the manometer is such that the pressure readings are proportional to the vertical displacement of a plunger in a chamber of oil. Table 2 gives calibration data for the draft gauge used for measuring velocity pressures.

To convert Pitot tube readings into air velocities the density of the air at the Pitot tube must be known.

TABLE 2.—CALIBRATION DATA FOR THE DRAFT GAUGE USED FOR MEASURING VELOCITY PRESSURE IN UNIT HEATER TESTS

ELLISON GAUGE READING INCHES OF WATER	STANDARD GAUGE READING INCHES OF WATER	ERROR
0.130	0.1314	−0.0014
0.146	0.1458	+0.0002
0.183	0.1810	+0.0020
0.186	0.1866	−0.0006
0.198	0.2000	−0.0020
0.224	0.2210	+0.0030
0.349	0.3522	−0.0032
0.408	0.4066	+0.0014
0.471	0.4737	−0.0027
0.507	0.5081	−0.0011
0.554	0.5527	+0.0013

This means that the temperature of the air must be known. Temperature readings were made simultaneously with Pitot tube readings by means of a thermocouple fastened to the side of the Pitot tube. The average temperature thus obtained was used in computing the air densities. Excepting tests numbers 1 to 6 the vapor content of the air was also considered. In Fig. 7 are typical curves showing how the temperature and velocity pressure varied across the duct.

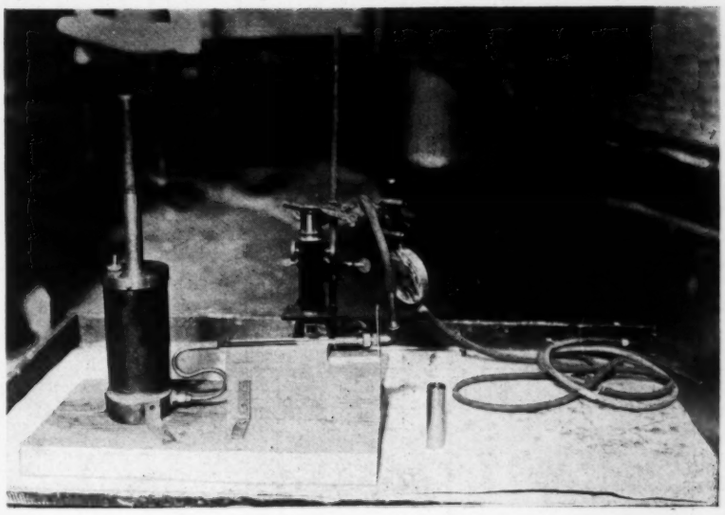


FIG. 6.—MANOMETER USED FOR CALIBRATING DRAFT GAUGES

The air volume output by both methods has been reduced to a standard density of 0.0749 lb. per cu. ft. This density corresponds to dry air at 70 deg. Fahr. and 29.92 in. heating pressure.

Assuming the Pitot tube method as a standard, percentages have been calculated showing the amount the condensate method is high or low. Glancing at these figures it is quite apparent that the two methods agree fairly well. In fact the agreement is so close that the writer finds himself in the paradoxical situation of having to explain a fact which from a rational viewpoint is axiomatic and yet so contrary to the experience of a majority as to be seriously questioned. The writer is referring to the almost universal conception that volumes obtained by the Pitot tube method are consistently much higher than volumes obtained indirectly by means of the condensate method. The thought is sometimes expressed by saying that the results by the condensate method are conservative.

If we may come immediately to some conclusion of practical value it is sufficient to say that these tests show that for any engineering purpose the difference between the air volume outputs obtained by the two methods is negligible. This statement, however, does not tacitly condone slipshod practices.

Returning to the percentages of error, the average amount by which the condensate method was low, using thermometers, was 1.3 per cent; using thermocouples, 0.11 per cent. The percentages range from — 2.00 per cent to + 2.99 per cent, using thermometers; and from — 3.20 per cent to + 2.31 per cent, using thermocouples. The larger negative values were obtained at low air velocities and the higher positive errors were obtained at the higher velocities. At first thought this would seem to indicate a systematic error in using the Pitot tube, especially in measuring velocity pressures, since the possibility for error becomes greater at low velocities. On the other hand, the final air temperatures increase as the air velocity decreases. It is quite probable that the temperature readings are in error for the same reason that we may suspect error in the Pitot tube. This is especially true with respect to the corrections which were added to final temperatures, since as was

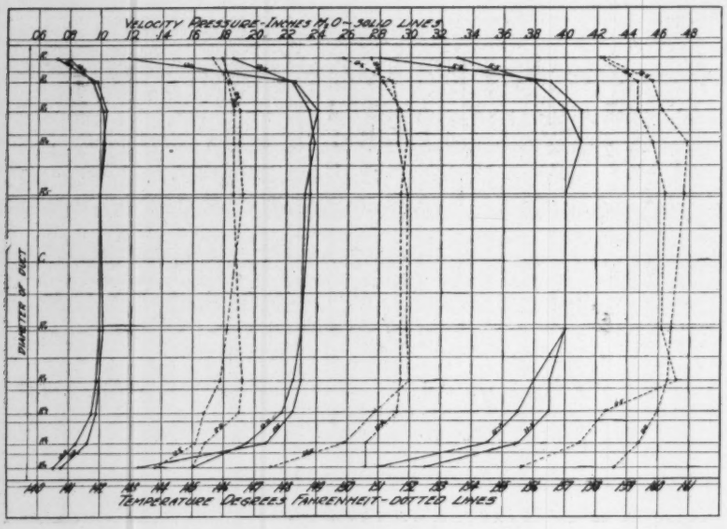

VELOCITY PRESSURE-INCHES H2O-SOLID LINES
06 08 10 12 14 16 18 20 22 24 26 28 30 32 34 36 38 40 42 44 46 48
DIAMETER OF DUCT
TEMPERATURE DEGREES FAHRENHEIT-DOTTED LINES

FIG. 7.—CURVES SHOWING VARIATION OF TEMPERATURE AND VELOCITY PRESSURES ACROSS DUCT

previously pointed out, these corrections were estimated on the basis of still air conditions on both sides of the insulation.

The variation of the percentage differences with respect to air velocity would have disappeared if the corrections applied to final temperatures had been a little greater at the low velocities and a little less at the high velocities.

There are many factors involved. For instance, it is stated in paragraph 83, Appendix 1, of the CODE OF MINIMUM REQUIREMENTS FOR THE HEATING AND VENTILATION OF BUILDINGS, that the specific heat of air "varies from 0.2375 to 0.2430 as determined by various investigators." These two values differ by approximately 2.3 per cent, which is more than enough to materially change the character of the results of the tests under consideration.

In regard to the measurement of air temperatures, thermocouples are to be preferred to thermometers on account of thermocouples being less influenced by radiation and air velocity effects, and also on account of their being adaptable to unusual circumstances.

The Pitot tube method has been considered the standard for determining air volumes. The Pitot tube is fundamentally accurate in theory and when properly designed it has proven accurate in practice. It would seem logical, therefore, to include the use of this instrument in a code for testing unit heaters. But since the testing of unit heaters includes other observations from which it is possible to calculate the air volume output with sufficient accuracy, it seems unnecessary and superfluous to include the Pitot tube or any other flow meter device such as the orifice or venturi tube.

The condensate method requires fewer instruments, less labor in testing and in making calculations, and a smaller amount of floor space for the set-up.

A. S. H. & V. E. Performance Test Code for Steam Heating Solid Fuel Boilers[1]

Code No 3[2]

Edition of June, 1929

1. **The object** of this code is to specify the tests to be conducted and to provide a standard method for conducting and reporting tests to determine the efficiencies and performance of the boiler.

2. **Chimneys.**—Tests may be conducted with either natural draft or induced draft. The boiler shall preferably be attached to a chimney of dimensions nearest to those cataloged by the manufacturer. If a proper sized chimney is not available and therefore induced draft or a larger chimney is used, means must be provided for reducing the draft so that the draft at no time exceeds that which would have been obtained with chimney as specified by the manufacturer.

3. **Boiler Set-Up.**—The boiler tested shall be a standard stock boiler. It shall be assembled in accordance with the manufacturer's directions, care being taken that all openings which would normally be closed (when manufacturer's directions are followed) in practice are closed with the proper type of cement or putty.

4. **If the boiler** is provided with an insulated jacket, this jacket shall be in place during tests. If boiler is uninsulated, all parts which would normally be covered in practice shall be covered with a coating of asbestos cement 1½ in. thick or equivalent insulation, or the boiler may be tested without covering and correction applied as explained in Par. 40 (Item *n*) under Computations. Parts of boiler which are not insulated in practice shall not be insulated during tests and shall not be considered as external boiler surface.

ARRANGEMENT OF APPARATUS

5. **The general arrangement** of apparatus shall be as shown in Fig. 1. Minimum apparatus for conducting tests shall be the following:

(A)—SCALES for weighing coal and residue

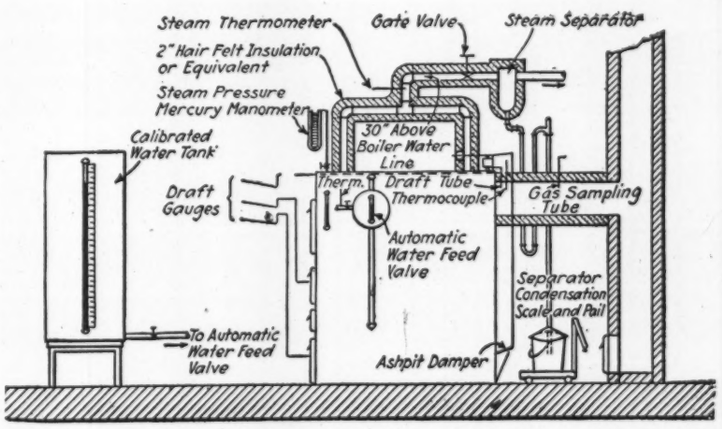


FIG. 1—ARRANGEMENT OF TEST APPARATUS

[1] Code prepared by F. B. Howell, *Chairman*, R. V. Frost, T. E. Langvoit, Carl H. Flink and ex-officios, Edwin W. Smith and Frederick W. Herendeen; adopted as Steam Heating Boiler Testing Code by the *National Boiler and Radiator Manufacturers' Association.*

[2] Adopted at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Bigwin Inn, Lake-of-Bays, Ontario, Canada, June, 1929.

SHEET NO. _____ OF _____ SHEETS

BOILER ROOM LOG SHEET—NEW FIRE METHOD

		DESIRED OUTPUT
BOILER	Name and Number	
	Description	Sq. Ft.
		% of Rating
	Manufactured by	Lbs. of Water
	Object of Test	

			POUNDS
Date of Test		Fuel Capacity at ____ ins. fuel depth	
Serial No. ____ Test No. ____		Fuel Available ____ % of above capacity)	
Fuel { Kind		a. Wood used in starting fire	
Fuel { Size		b. Fuel equivalent of wood (a)×0.4	
Chimney: Size ____ (dia.) Height (ft.)		c. Fuel fired during test	
Induced Draft		d. Total fuel as fired (b)+(c)	
FUEL ANALYSIS		e. Moisture in fuel % lbs.	
Proximate %	Ultimate %	f. Total dry fuel fired (d)−(e)	
Volatile Matter	Carbon	g. Total residue and cleanings	
Fixed Carbon	Hydrogen	h. Combustible burned (f)−(g)	
Ash	Nitrogen	i. Net dry fuel burned (h)+(l)	
Sulphur	Oxygen	j. Dry fuel burned per hour (l)÷hrs.	
	Sulphur	k. Dry fuel burned/hour/sq. ft. grate (j)÷G.A.	
Calorific Value ____ B.t.u. per lb.		l. Combustible per pound of fuel from Analysis	
Steam Outlets Used: No. ____ Size (ins.)		Cleanings from Ashpit: Total	
Location		Ash (lb.) Clinker (lb.)	
Grate Area* (sq. ft.) ____ Heating Surface (sq. ft.)		Unburned (lb.) Slate (lb.)	
*Fire box at Grate Level	External Surface (sq. ft.)	Clinker from Firebox	

	Hr. of Test	TIME Hrs.	TIME Min.	WATER (LBS.) Reading on Tank Gauge	WATER (LBS.) Reading on Tank Gauge	WATER (LBS.) Evap. between readings	WATER (LBS.) Evap. per hour	WATER (LBS.) Total Evap. from start of Test	DRAFTS (ins. H_2O) In smoke Outlet	In Fire-box	In ash pit				Barom. Pressure Ins. Hg.	Steam Pressure Lbs. per Sq. In.	TEMPERATURES °F Feed Water	Steam	In Smoke Outlet	Boiler Room	Out-side Air
Total																					
Average																					

			POUNDS
m. Actual evaporation x factor of evaporation			
n. Corrections	Heat loss from uncovered boiler		
o.	Heat loss from uncovered piping		
t. Boiler Eff.	%	p. Total Equivalent Evaporation (m)+(n)+(o)	
		q. Equivalent Evaporation per hr. (p)÷hrs.	
u. B & G Eff.	%	r. Output in Sq. Ft. of Radiation (q)×970.4÷240	
		s.	

Other side for additional remarks, fuel weights, residue and gas anal

SHEET NO _____ OF _____ SHEETS

BOILER ROOM LOG SHEET—CONTINUOUS FIRE METHOD

		DESIRED OUTPUT
BOILER	Name and Number	
	Description	Sq. Ft.
		% of Rating
	Manufactured by	Lbs. of Water
	Object of Test	

			POUNDS
Date of Test		Fuel Depth—Beginning and end of test inches	
Serial No. ____ Test No. ____		a. Total fuel as fired	
Fuel { Kind		b. Moisture in fuel % lbs.	
Fuel { Size		c. Total dry fuel fired (a)−(b)	
Chimney: Size ____ (dia.) Height (ft.)		d. Cleanings from ashpit	
Induced Draft		e. Clinker from fire box	
FUEL ANALYSIS		f. Total cleanings (d)+(e)	
Proximate %	Ultimate %	g. Combustible in total cleanings % lbs.	
Volatile Matter	Carbon	h. Fuel equivalent of combustible in cleanings (g)×14500 cal. value	
Fixed Carbon	Hydrogen	i. Net dry fuel burned (c)−(h)	
Ash	Nitrogen	j. Dry fuel burned per hour (i)÷hrs.	
Sulphur	Oxygen	k. Dry fuel burned per hour per sq. ft. grate (j)÷G.A.	
	Sulphur		
Calorific Value ____ B.t.u. per lb.			
Steam Outlets Used: No. ____ Size (ins.)			
Location			
Grate Area* (sq. ft.) ____ Heating Surface (sq. ft.)			
*Fire box at Grate Level	External Surface (sq. ft.)		

	Hr. of Test	TIME Hrs.	TIME Min.	WATER (LBS.) Reading on Tank Gauge	WATER (LBS.) Reading on Tank Gauge	WATER (LBS.) Evap. between readings	WATER (LBS.) Evap. per hour	WATER (LBS.) Total Evap. from start of Test	DRAFTS (ins. H_2O) In smoke Outlet	In Fire-box	In ash pit				Barom. Pressure Ins. Hg.	Steam Pressure Lbs. per Sq. In.	TEMPERATURES °F Feed Water	Steam	In Smoke Outlet	Boiler Room	Out-side Air
Total																					
Average																					

			POUNDS
m. Actual evaporation x factor of evaporation			
n. Corrections	Heat loss from uncovered boiler		
o	Heat loss from uncovered piping		
t. Boiler Eff	%	p. Total Equivalent Evaporation (m)+(n)+(o)	
		q. Equivalent Evaporation per hr. (p)÷hrs.	
u B & G Eff.	%	r. Output in Sq. Ft. of Radiation (q)×970.4÷240	
		s.	

Other side for additional remarks, fuel weights, residue and gas anal

(B)—Calibrated weighing tank or other means for determining water evaporated

(C)—Thermometers for determining feed water and steam temperature

(D)—Mercury manometer for determining steam pressure

(E)—Draft gauges for determining ash pit, fire box, and smoke outlet draft

(F)—Pyrometer for determining temperature of gases leaving boiler

(G)—Recording or indicating CO_2 and CO apparatus and means for checking recording apparatus

(H)—Scale for weighing condensation obtained from separator

(I)—Barometer or barograph for determining atmospheric pressure during test.

6. **Location of Instruments.**—The feed water thermometer shall be inserted in a thermometer well so located as to read the true temperature of the water entering the boiler. The steam temperature thermometer shall be inserted directly into the steam in the flow outlet.

7. **The mercury manometer** shall be connected with the steam space of the boiler.

8. **A draft gauge** shall be connected to an open end tube set at right angles to the flow of gas in each of the following: ash pit, fire box, and smoke outlet collar. To prevent influence upon gas samples, draft reading, or gas temperature reading, choke dampers between flues and smoke outlet collar shall be left in wide open position throughout all tests. Cold air checks between smoke outlet collar and flues shall be thoroughly sealed during tests. Air leakage around draft tubes, thermocouples and gas sampling tubes shall be prevented by using asbestos rope or other suitable means packed around the tubes.

9. **The temperature** of gases leaving the boiler shall be taken by means of an exposed thermocouple located as shown on Fig. 2. The smoke hood and smoke pipe shall be covered with two inches of asbestos cement to a distance one foot beyond the thermocouple.

10. **The gas** sampling tube shall be a single open end tube in position corresponding to gas temperature thermocouple and shall be placed in the smoke pipe within one foot of the gas temperature thermocouple as shown on Figure 2.

11. **Accuracy of Instruments.**—All instruments shall be calibrated to insure accuracy.

12. **Fuels to be Used.**—For anthracite boilers the fuel used shall be the commercial size, best suited to the boiler, as specified by the manufacturer. For soft coal boilers the coal used shall be a free burning 3 x 2 in. lump coal unless the boiler performs more satisfactorily with caking coal. If caking coal is used it shall be of approximately 3 x 2 in. lump size. For coke boilers the coke used shall be by-product or gas coke of commercial size, best suited to the boiler.

13. **If the boiler** is designed for a special solid fuel, tests shall be conducted with this fuel. In all cases the characteristics, kind, and size of fuel used in making tests shall be recorded on the test form. Sizes of fuel used shall be according to classification of *American Society of Mechanical Engineers,* as given in Table 1.

Table 1. Names and Sizes of Various Fuels

Name and Size	Screen Through	Screen Over
Domestic Hard Coals		
Pea	¾″	½″
Chestnut or Nut	1¼″	¾″
Stove or Range	1¾″	1¼″
Egg—in the East	2½″	1¾″
Large Egg—Chicago	4″	2¾″
Small Egg—Chicago	2¾″	2″
Broken or Grate	4″	2½″
Domestic By-Product Coke		
Egg	3″	2½″
Large Stove	2½″	2″
Small Stove	2″	1½″
Nut	1½″	¾″
Pea	¾″	½″
Bituminous Coal		
Duff	⅛″	....
No. 3 Nut	1¼″	¾″
No. 2 Nut	2″	1¼″
No. 1 Domestic Nut	3″	1½″ or 2″
No. 4 Washed	¾″	¼″
No. 3 Washed Chestnut	1¼″	¾″
No. 2 Washed Stove	2″	1¼″
No. 1 Washed Egg	3″	2″
No. 3 Roller Screened Nut	1½″	1″
No. 2 Roller Screened Nut	2″	1½″
No. 1 Roller Screened Nut	3½″	2″
Egg	6″	3″
Lump or Block	....	6″
Run-of-Mine: fine and large lumps.		
Screenings: usually smallest sizes.		
Pocahontas Smokeless: Generally sized as: Nut, Egg, Lump, and Mine Run.		
Cannel Coal: For fireplaces—Hand Picked Lump. For stoves—Egg.		

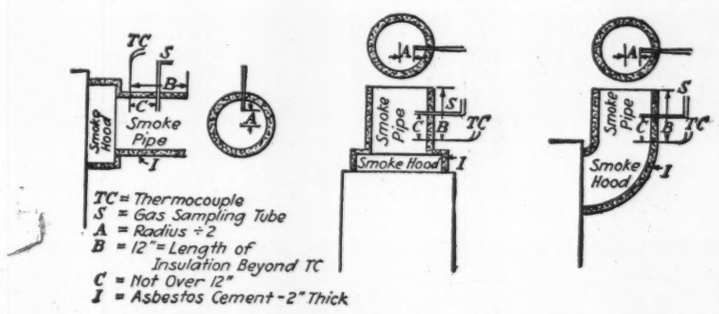


Fig. 2—Location of Gas Temperature Thermocouple and Gas Sampling Tube

TESTS TO BE CONDUCTED

14. **Five tests** shall be conducted at 2 lb. steam gauge pressure—one shall be at drive output and the other four at various output rates to show the entire operating range of the boiler. The manufacturer shall determine and designate the output rates and the operating range of the boiler. No test data or boiler information shall be assumed for an output beyond the drive test. The drive test shall be considered the boiler's maximum output. No tests shall be reported in which the water collected from the separator is greater than 2 per cent of the water fed to the boiler in excess of the condensation resulting from the normal heat loss of the separator and piping. Correction for the water collected must be applied to the water evaporated in order to include in the output only the heat actually absorbed to heat the water from feed water temperature to steam temperature.

15. **Outlet Piping.**—The number and size of outlets shall be as cataloged by the manufacturer. Outlets shall be full size to a height 30 in. above the boiler's normal water line. An effective separator shall be placed in the steam outlet as near the boiler as practicable. The separator shall be open to the atmosphere through a water seal permitting any water separated from the steam to drain immediately into a container for weighing. Re-evaporation of water drained from separator shall be prevented by use of a suitable cover on the container.

16. **A gate valve,** full size of flow piping, shall be placed

between boiler and separator for the purpose of maintaining a constant steam pressure in the boiler at various test output rates. All piping up to and including separator shall be insulated with 2 in. of hair felt, or equivalent.

17. **Outlet piping** shall pitch away from the risers and away from the separator.

18. **For all boilers** up to 3,000 sq. ft. of steam radiation, outlet to chimney shall be of size cataloged by manufacturer and shall be air tight and as short and direct as possible, preference being given to long radius and 45 deg. instead of 90 deg. bends. The bend at entrance to chimney shall be carefully made, care being taken that the pipe does not project beyond inside of chimney and that no leakage of air occurs.

WOOD				RESIDUE			
TIME FIRED	GROSS	TARE	NET	TIME WEIGHED	GROSS	TARE	NET
COAL							

HEATING BOILER TEST REPORT—SOLID FUEL BURNING

| | | |

The boiler shall then be drained and refilled with clean water.

21. Duration of Test.—Each test shall be conducted at the discretion of the engineer until there has been burned in pounds per square foot of grate not less than 250 in the continuous fire method or 150 in the new fire method, or until the error in fuel and water determination shall not exceed 2 per cent.

22. Calorific Tests and Analysis of Fuels and Refuse.—The quality of the fuel and of the ash and refuse shall be determined by calorific tests and analysis of samples. Directions for obtaining samples of the fuel and making these tests and analyses shall be the recommendations in the *A. S. M. E. Test Code for Solid Fuels*—April 1927.

23. A representative sample of the fuel shall be taken during the test and shall be kept in an air tight container pending determination of moisture.

STARTING AND STOPPING OF TEST

24. New Fire Method.—New fire method of starting and stopping test may be used on anthracite or coke boilers. A preliminary fire using wood or coal shall be made and the boiler shall be operated until the water in the boiler has reached steam temperature and the boiler insulation normal operating temperature. During this preliminary operation the water in the gauge glass shall be kept at the normal boiler water line. The fire shall then be quickly removed from the boiler, the ash pit cleaned, the flues thoroughly cleaned, the quantity of water in tank noted, and the test started by placing a charge of kindling wood on the grate (9 lb. for each sq. ft. of grate) igniting same and firing 33½ per cent of the **Fuel Charge (Available Fuel).** A second amount of fuel equal to the **Fuel Charge (Available Fuel)** shall be fired within the first half hour of test. No firing or attention to fuel shall be permitted after this first half hour period until the time of adding the next fuel charge.

25. When test has been conducted for such a period as specified under *Duration of Tests* (par. 21), and the fuel has burned down to that point where the output can no longer be maintained by operation of the dampers, the test shall be ended as follows:

(A)—THE GRATES shall be shaken to remove ash from the fire as if another charge of fuel were to be added

(B)—THE CLEANINGS shall be quickly removed from the ash pit and weighed

(C)—THE RESIDUE within the fire box shall be quickly quenched with just sufficient water to put out the fire. (The quantity of water used shall be recorded.)

(D)—THE ENTIRE residue of the fire box shall be removed and weighed, and shall be kept separate from the cleanings. The water level at the end of the test shall be the same as at the beginning of test.

26. Continuous Fire Method.—The continuous fire method may be used for all boiler tests. The boiler shall be operated under test conditions for at least one firing period and not less than one hour before the starting of test. A preliminary fire shall be made and the boiler operated until the water has reached steam temperature, and the boiler insulation normal operating temperature. The flues shall then be thoroughly cleaned. The fire shall then be burned low, thoroughly cleaned, and the remaining live fuel spread evenly over the grate as the foundation for the first to test fuel charge. If the manufacturer provides firing directions these directions shall be followed.

27. The thickness of the fuel bed shall be quickly measured and the condition of the fuel bed noted. The height of water in boiler and feed tank shall be recorded. Tests shall start at the time of making these observations. The first weighed charge of fuel (see Par. 29) shall then be fired. If firing directions are specified by manufacturer, same shall be followed. The ash pit shall be thoroughly cleaned immediately and the test allowed to proceed.

28. At the end of the test fire shall be burned low and cleaned so as to leave the fuel bed in the same condition and thickness as at the start of the test. When this condition is reached the water level in the boiler shall be at the same height as at the start and the test shall be ended at this time. The contents of the ash pit shall be quickly removed and weighed.

29. The Fuel Charge for ALL tests shall be:

(A)—The amount specified by the manufacturer, or

(B)—The amount determined by the engineer in charge of test as the maximum amount of fuel consistent with proper boiler performance.

30. The Fuel Charge (Available Fuel) used in computations to determine the *Average Interval between Firings* which is recorded on Form D, Item 17, shall be obtained as follows:

(A)—NEW FIRE METHOD: Divide the total quantity of fuel fired during test (but not including the wood or coal used as a preliminary charge) by the number of firings during test.

(B)—CONTINUOUS FIRE METHOD: Divide the total quantity of fuel fired during test by the number of firings during test.

31. Attention to fire shall be considered as any one or more of the following operations and in all tests burning anthracite or coke of nut or larger size there shall be no shaking of grates or other attention to fire except at time of recharging boiler.

(A)—FIRING
(B)—STOKING
(C)—LEVELING
(D)—SHAKING GRATE

32. The average interval between the end of any attention period and the beginning of the next shall be stated in the report of test.

33. All dampers controlling the primary and secondary air may be adjusted as required to produce most efficient combustion at any time during test.

34. The following readings shall be recorded at regular intervals not greater than 15 minutes throughout test:

(A)—WATER evaporated
(B)—TEMPERATURE of feed water
(C)—STEAM temperature
(D)—STEAM pressure
(E)—DRAFTS in ash pit, fire box and smoke outlet
(F)—TEMPERATURE of gases leaving boiler
(G)—WEIGHT of water from separator

35. Following readings shall be recorded every hour:

(A)—BOILER room temperature
(B)—OUTSIDE temperature
(C)—BAROMETRIC pressure

36. In all tests using bituminous coal, any smoke observations shall be recorded every 15 seconds according to Ringelman Chart numbers throughout two complete firing periods.

TEST FORMS

37. Results of tests shall be recorded on test forms as follows:

(A)—BOILER ROOM LOG SHEET—Form A (for new fire method) or Form B (for continuous fire method) shall be used for recording all readings during test.

(B)—THE BACK OF THE BOILER ROOM LOG SHEET—Form C shall be used for recording weights of fuel, residue, gas analysis (if Orsat Apparatus only is used), or for recording check of recording gas analysis apparatus with Orsat Apparatus.

38. Results of all tests on one boiler shall be reported on Form D.

39. The following results referred to boiler output in

square feet of steam radiation, taken from Form D shall be shown by curves on Form E:

(A)—Boiler and grate efficiency
(B)—Temperature of gases leaving boiler
(C)—Average firing interval
(D)—Average attention interval
(E)—Draft difference (smoke outlet minus ash pit).

COMPUTATIONS

40. Form A.

Item *(b)* = The factor 0.4 is the weight of coal equivalent in calorific value to 1 lb. of wood.

Item *(n)* = (External Boiler Surface) × (1.8) × ($T_s - T_r$) × (hrs.) × 1/971.7.
1.8 = (Coef. of uncovered boiler surface—coef. of covered boiler surface)
= B.t.u./sq. ft./hr. deg. fahr. diff. of surface and room.
T_s = Surface temperature = steam temperature.
T_r = Room temperature.
hrs. = Duration of test.

Item *(o)* = Surface × 2.4 × ($T_s - T_r$) × (hrs.) × (1/971.7).
(Surface) = External surface of pipe in sq. ft.
2.4 = Average coefficient for uncovered pipe.
T_s = Steam temperature.
T_r = Room temperature.
hrs. = Duration of test.

$$\text{Item } (t) = \frac{\text{Item } (p) \times 971.7}{\text{Item } (i) \times (\text{Calorific value})} \times 100$$

$$\text{Item } (u) = \frac{\text{Item } (p) \times 971.7}{[\text{Item } (i) + \text{Fuel through grate}] \times (\text{Calorific value})} \times 100$$

41. Form B.

Item *(h)* = 14,600 = Calorific value of 1 lb. carbon.
It is assumed that the combustible in ash is carbon.

$$\text{Item } (u) = \frac{\text{Item } (p) \times 971.7 \times 100}{\text{Item } (c) \times \text{Calorific value}}$$

Item *(n)*, *(o)* and *(t)* same as shown under Form A.

42. Form D.

$$\text{Item } 8 = \frac{\text{Fuel through grate}}{\text{Dry fuel burned} + \text{fuel through grate}} \times 100 \text{ (New Fire Method)}$$

or

$$\frac{\text{Fuel through grate}}{\text{Dry fuel fired}} \times 100 \quad \text{(Continuous Fire Method)}$$

$$\text{Item } 9 = \frac{11\ CO_2 + 8\ O_2 + 7\ (CO + N_2)}{3\ (CO_2 + CO)} \times \frac{C \times 0.24 \times (T_f - T_r)}{\text{Calorific value}}$$

CO_2, O_2, CO and N are in o/o by volume obtained and
C is taken as per cent carbon by weight per pound of fuel.
T_f = Temperature of gases leaving boiler.
T_r = Room temperature.

$$\text{Item } 10 = \frac{CO}{CO_2 + CO} \times C \times \frac{10{,}160}{\text{Cal. value}} =$$

CO_2, CO and C same as for Item 9.

$$\text{Item } 11 = \frac{\text{Surface of boiler} \times \text{Coef. of covered boiler surface} \times (T_s - T_r) \times 100}{\text{Fuel burned per hour} \times \text{Calorific value}}$$

$$\text{Item } 17 = \frac{\text{Fuel charge} \times (\text{boiler and grate efficiency}) \times 12{,}500}{\text{Output} \times 240}$$

$$\text{Item } 18 = \frac{\text{Hours}}{\text{Times given attention}}$$

$$\text{Item } 19 = \frac{\text{Weight of clinker removed}}{\text{Dry fuel burned} + \text{Fuel through grate}} \times 100 \text{ (New Fire Method)}$$

or

$$\frac{\text{Weight of clinker removed}}{\text{Dry fuel fired}} \times 100 \text{ (Continuous Fire Method)}$$

DEFINITION OF TERMS USED IN THE CODE

43. Boiler Efficiency is the percentage of the heat in the net dry fuel burned delivered at the boiler nozzle when the boiler is normally insulated.

44. Boiler and Grate Efficiency (Overall Efficiency) is the percentage of the heat in the net dry fuel burned plus the heat in the fuel fallen through the grate delivered at the boiler nozzle when the boiler is normally insulated.

45. Fuel Charge (Available Fuel) is the maximum amount of fuel that can actually be fired at one time and can be burned with proper boiler performance between firing intervals. It must be demonstrated during tests that the quantity cataloged by the manufacturer as **Fuel Charge (Available Fuel)** can actually be fired at one time upon a fuel bed that is sufficiently deep to properly ignite the **Fuel Charge.** The following values for weight of fuel per cu. ft. shall be used for catalogue purposes:

Anthracite	52.5
Bituminous	45.0
Coke	28.0

46. Fuel Capacity is the weight of fuel that must be left in the boiler to kindle a fuel charge plus the weight of fuel which can be fired at one time. When specifying fuel capacity, it is necessary to specify the kind of fuel. Fuel capacity is not the total weight of fuel that can be shoveled into the fuel and combustion space because there must be sufficient furnace volume in addition to that occupied by the fuel to permit the escape of the gases leaving the fuel.

47. Calorific Value is the number of B.t.u. per pound of dry fuel as determined in a bomb calorimeter.

48. Clinker.—(Item 19, Form D) is the weight of clinkers which must be removed through fire or clinker doors. It does not include any clinker that drops into ash pit either during normal operation or when grates are shaken.

49. Residue.—(Item *g*, Form A) is the weight of the contents of the firebox when ending a test in the new fire method.

50. Cleanings.—(Item *g*, Form A) is the weight of material that drops through or is shaken through the grate into the ash pit.

51. Combustible per Pound of Fuel from Analysis.—(Item *l*, Form A) equals one minus the weight of ash obtained from the analysis of the dry fuel.

52. External Surface (Form A and Form B) is the measured surface of the boiler which is normally insulated in an installation.

53. Radiation from Covered Boiler (Item *ll*, Form D) is the heat lost through insulated surfaces of the boiler. It is not included in boiler output but is shown on Form D to give the manufacturer an idea of the extent of this loss.

54. Heat Loss from Uncovered Boiler (Item *n*, Form A and B) is the heat lost from that part of the boiler surface not insulated during test which could be saved by the normal insulation of the boiler in practice and which would therefore appear as added output.

55. Heat Loss from Uncovered Piping (Item *o*, Form A and Form B) is the heat that is lost through uncovered piping (if condensation resulting drains back to the boiler and is re-evaporated) because the surfaces thereof are not normally insulated as specified in code. This correction is required only when piping cannot be insulated.

56. Coefficient of Covered Boiler Surface is the quantity of heat expressed in B.t.u. transmitted from the steam and water in the boiler to the boiler room per sq. ft. of normal insulation per hour per degree difference in temperature between the steam and the boiler room.

57. Actual Evaporation is the actual weight of water heated from the average feed temperature to average steam temperature and evaporated into steam during a test.

58. Factor of Evaporation is the total number of B.t.u. absorbed per pound of water evaporated divided by 971.7.

59. Total Equivalent Evaporation (Item *p*, Form A and Form B) is the quantity of water which would have been evaporated from a feed water temperature of 212 deg. into steam at 212 deg. by the corrected heat output delivered at the boiler nozzle.

60. Boiler Output is the quantity of heat available at the boiler nozzle with the boiler normally insulated.

61. The Square Foot of Steam Radiation is the heat output of 240 B.t.u. per hour.

62. A Damper is a device used to control the flow of air or gases in a boiler.

63. A Choke Damper is a damper placed within the gas passage.

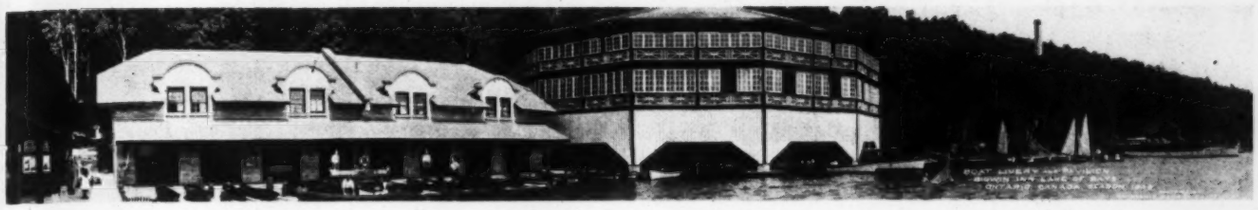
64. A Check Damper is a device for admitting air from the boiler room to the gas passage or smoke pipe.

PRESIDENT'S PAGE

There are hundreds of splendid engineers who are not in our ranks and who ought to be with us in order to advance the aims for which the Society was organized.

Our Society can only be maintained in a healthy condition by a constant infusion of new blood, of good quality, and this infusion must come from the younger men in the profession.

Have you endeavored to interest a younger man in the activities of the Society? If not, will you do so?



THE PAVILION AT BIGWIN INN

Proceedings of Semi-Annual Meeting

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

Bigwin Inn, Lake-of Bays, Ontario, Canada

June 26 to 28, 1929

THE Ontario Chapter of the A. S. H. V. E. may well be proud of the fact that the Semi-Annual Meeting, 1929, held at Bigwin Inn, Lake-of-Bays, Ontario, was not only the first gathering of the Society outside of the United States but was also one of the best attended sessions held in the 35 years of the organization's history.

The Canadian members acted as hosts to 380 ladies, members and guests from all parts of the United States and the Dominion. They found Bigwin Inn a revelation and the hospitality and spirit of their hosts unsurpassed.

Technical sessions were interesting and well attended. Decisive action was taken on several Codes. Discussions were well prepared and delivered while the facilities for sport and other entertainment will be remembered for years to come.

Definite forward progress was made in the solution of the boiler testing and rating problem, a code for testing and rating unit heaters is in prospect, and regulations for the heating and ventilating of garages were adopted so that the Semi-Annual Meeting 1929 can be put on record as one of progressive action.

First Session—Wednesday, June 26, 9:30 a. m.

The Semi-Annual Meeting of the Society was called to order by President Lewis on Wednesday, June 26 at Bigwin Inn, Lake-of-Bays, Ontario, Canada and M. Barry Watson, president of the Ontario Chapter welcomed the members to Canada for the first meeting of the Society, outside of the United States. In response, President Lewis voiced the appreciation of the Society for the splendid work of the Ontario Chapter members and said that it was fitting that a scientific Society should not be restrained by national borders. President Lewis said that science is international and men engaged in the same profession are brothers no matter where found.

The first paper presented was entitled, Instruments for the Measurement of Air Velocity, by J. H. Parkin, Toronto. In the absence of the author, the paper was presented in abstract by J. J. Aeberly, Chicago (Complete text of this paper appeared in the June JOURNAL, p. 145).

Dr. E. Vernon Hill, Chicago, stated that the paper gave a fine description of instruments and the methods of use. W. A. Rowe, Detroit, stated that the successful use of the anemometer depended upon the accuracy of its calibration as well as upon proper technique in its use. He also pointed out that recent developments would soon bring forth means for measuring extremely low air velocities by resistance wire instead of with an anemometer.

Prof. F. E. Giesecke, Texas, told of a problem requiring uniform air distribution across the hoods in a chemical laboratory and he outlined the method of studying the air distribution which was by arranging wires across the face of the hood at every foot and suspending paper flags at a distance of 4 to 6 in. from the wire so that the inclination of the flags showed the air velocity at different points.

Prof. A. P. Kratz, Urbana, Ill., stated that from results obtained in air measurement studies with the anemometer at the University of Illinois, it had been found that calibration of the instrument must be made under the same conditions as would exist when it was used.

F. C. Houghten, Pittsburgh, stated that plans were under way to make a detailed study of the subject of air flow and pointed out that the preliminary work indicated that commonly accepted methods of air measurement were in error from 25 to 40%.

W. H. Carrier, Newark, N. J., emphasized the importance of the contemplated study of the problem of air measurements and thought that a code should be prepared in the laboratory rather than by a Technical Advisory Committee.

E. D. Milener, New York City, said that air volumes passing through furnaces and ducts were matters of concern to the *American Gas Association* and related that accurate measurements were obtained by using the Thomas meter.

President Lewis introduced Prof. A. P. Kratz,

University of Illinois, who presented a paper entitled, Analysis of the Overall Efficiency of a Residence Heated by Warm Air (published June 1929 JOURNAL, p. 33).

John Howatt, Chicago, inquired about the term "coal fired" and pointed out that it would be difficult to get an overall heating efficiency of 95 per cent when it was common to find 30 per cent combustible in the ordinary ash test. He also believed that houses with chimneys on the outside wall were more frequently built than those where the chimney is inside and that less heat would be recovered in the average residence than was indicated by the research residence tests. Inquiry was also made by Mr. Howatt, about the relation of heating costs when an open fireplace was used.

Prof. Kratz stated that all calculations of efficiency and performance were made on the basis of coal burning and said that combustible losses in the ash pit were negligible because grate efficiencies range from 94 to 97 per cent. He explained that the heat regained from the chimney is not large but that it comes mostly from the warm air stacks and from the smoke pipe and furnace casing. No tests with open fireplaces have been made but the author indicated that they were in prospect.

R. V. Frost, Norristown, Pa., congratulated the author on bringing out the point of high overall efficiencies and stated that the assumed low efficiencies in house heating practice were over emphasized. He cited results of recent tests with domestic stokers indicating that boiler efficiencies based on cast iron and steel units burning anthracite coal were frequently as high as 85 per cent.

Homer Linn, Chicago, inquired about the floor and ceiling temperatures during the tests.

Prof. Kratz explained that on a zero day in the first floor rooms, the temperature differences between floor and ceiling averaged 12 to 15 deg. and he pointed out that temperature differences at the two points mentioned were dependent on outdoor temperatures. On the second floor the average temperature between floor and ceiling points was slightly higher or approximately 16 deg. and on the third floor from 17 to 18 deg. This temperature differential was found to be common with practically all types of house heating apparatus. Auxiliary air above the fire was found to be of little value in house heating apparatus because of the comparatively low combustion rates used.

Prof. L. M. Arkley, Ontario, asked about uniform distribution of heat, comparing the warm air, steam and hot water systems. Prof. Kratz told about tests in the research residence where temperature measurements were taken at the breathing line of various rooms, with communicating doors both open and closed, and stated that the maximum differences did not exceed 3 deg. He said that no tests had been run, except with warm air furnaces, but felt that the overall efficiency could not be materially different.

T. F. McCoy, Boston, asked if the air supply was from indoors or out-of-doors and Prof. Kratz stated that the air was all recirculated.

Prof. G. L. Larson, Madison, Wis., asked if the tests were conducted under maximum efficiency conditions. Prof. Kratz stated that curves were plotted over the entire range of outdoor temperatures but that a tendency to overheat the house was noticed when the outdoor temperature was above 50 deg.

H. Gurney, Ontario, asked about the heat loss through cellar, walls and windows and the author pointed out that in an accurately designed warm air furnace system, the air circulation is so rapid through the casing that the heat loss from the basement is not material.

F. W. Johnson, Detroit, questioned the 175 deg. outlet temperature used in making computations and Prof. Kratz stated that all present day codes were based on 175 deg. as the basic temperature for design and pointed out that in the actual performance of the plant, register temperatures in the neighborhood of 140 deg. result because of heat transmission losses.

C. W. Deland, Chicago, wanted to know whether it was difficult to heat a northwest room which had a large glass surface and Professor Kratz stated that the breathing line temperature during the first year test was 4 deg. lower than the average for the rest of the house. Increasing the size of the leader pipe had not remedied the trouble, but insulating the ceiling of the room brought the temperature up.

H. L. Walton, Detroit, wondered whether the actual fuel consumption, which was less than the calculated fuel requirement, might not be accounted for on the basis of the heat loss computations, which is not an exact science.

Professor Kratz pointed out that as a check on the accuracy of the heat loss calculations, the amount of heat in the coal fired compared very well with the overall efficiency as determined from the heat appearing in the flue gas, and as a second check it was noted that no room failed to heat by any large percentage of what had been calculated. Any material error in heat loss calculations would have been indicated in an individual room, as well as in the house as a unit, he explained.

The next speaker introduced by President Lewis was H. L. Walton, Detroit, who gave an abstract of the paper, Air Conditioning System of a Detroit Office Building, by H. L. Walton and L. L. Smith, which was published in the May 1929 Journal, p. 43. While outlining his paper, the author gave some rather interesting operating figures during summer operation. He stated that on June 19, a dry bulb temperature of 91 deg. and a relative humidity of 60 per cent was experienced, and that uniform temperature conditions existed throughout the various floors served by the air conditioning system. During another period it was noted that the effective temperatures maintained were from 66.5 deg. to 69.3 deg., and the variation had not been uncomfortable to the occupants. The speaker stated that it was believed that winter time operation would be more favorable for control of the temperatures in the rooms, as the amount of heat could be varied in different parts of the system, while in the summer time the only opportunity for control is at the dehumidifying unit, while the amount of air supplied

to any exposure supplies all floors on a riser and not just an individual floor.

L. A. Harding, Buffalo, was impressed with the fact that the authors, for the first time, had applied the term *effective temperature* to an actual installation, and he also complimented them on their summarizing the items often termed *fixed charges*. He commented on the ratio of refrigerating capacity as compared with building occupants and the customary practice in theatres, and inquired about the number of air changes used as compared with the practice in theatre work.

Two written discussions were presented by P. D. Close, Technical Secretary.

The comments of Thomas Chester, Pittsburgh, pointed out that the novelty of the air conditioning system described was the treatment of the 7th to 16th floors, and questioned whether the carrying of a relatively high static pressure on the system would overcome the effect of all leakage into the building through windows, doors, elevators, etc. He also suggested that the entering air be projected toward the windows to avoid cold drafts, and believed that better results might be obtained by the introduction of air through long, narrow grills in window boxes.

The written discussion of M. G. Harbula indicated that the writer was impressed with the collaboration between the architectural and mechanical designers, and the fact that the air distribution ducts were properly proportioned, and still met structural conditions. The writer believed that practical operation will demonstrate that the operating cost might be lowered in power consumption, cost of condensing water and cost of refrigerant.

James Govan, Toronto, speaking as an architect, emphasized the point that air conditioning in office buildings provided a problem in acoustics which deserved study, and air conditioning engineers, acoustical engineers and architects must provide methods of eliminating sound transmission through ducts.

S. R. Lewis, Chicago, called attention to two items which sometimes cause difficulties, first, the matter of temperature regulation in a number of comparatively small offices served by the cooling system, and second, the uncertain operation of several fans pulling through the heavy resistance of a dehumidifying unit and duct system.

Dr. E. V. Hill said that there are two points to which he should like to focus attention, first the matter of public acceptance of the completely air-conditioned modern building, and secondly, the effectual dissemination of facts regarding air conditioning among professional men and Society members. Every engineer, he states, should take the position that today he can create and maintain better air conditions indoors than nature provides outdoors. This is not imitating nature, but rather improving upon nature, according to Dr. Hill. It is not to be inferred that it is more healthful to live indoors than to live outdoors, but it is a fact that air indoors is better than is found in the city out-of-doors. The outdoor life is desirable and healthful on account of the sun effect, exercise and other things.

H. M. Nobis, Cleveland, inquired about the effect of eliminating radiation under windows, and also asked whether the maintenance of uniform air conditions in offices would be as effective in stimulating physical reactions as outdoor air. It was his belief that the main value of air conditioning installations was the elimination of dust, smoke and other objectionable products from city air.

President Lewis pointed out that the air conditioning engineer could furnish any condition of air cleanliness and humidity for any type of building, and the great need was to have medical men establish conditions which are most healthful for human beings.

S. S. Sanford, Detroit, mentioned a service building of the Detroit Edison Company which is heated with warm air introduced through ceiling diffusers with a small additional amount introduced through slots at the windows. Apparently, the quantity of air handled by this plant is insufficient to entirely overcome window drafts in severe weather, and it is noted that there are two places in the room where complaints originate, one, in a corner near an enclosed stairway and the other at the exposed opposite corner where excessive drafts are found. It is believed that a small amount of direct radiation under each window will correct the difficulty.

In reply to the various questions, Mr. Walton explained the reason for the low percentage assigned to fixed charges, also stating that the tonnage required for cooling the building was due to exposure rather than population; he told of the advantages anticipated from maintaining a plenum effect, and outlined the manner in which acoustical difficulties had been provided for, with the result that no objectional sound transmission from one office to another had been noted. The author explained that temperature regulation was considered one of the most critical points in designing the installation, and the temperature variations so far recorded had met the requirements of the occupants. It was explained that only such portions of the building as occupied by the owners had been air conditioned.

President Lewis next introduced Professor O'Bannon, author of the paper, Heat and Air Volume Output of Unit Heaters, which represents co-operative work between the A. S. H. V. E. Laboratory and the University of Kentucky. (Published this issue, p. 323.)

Chairman Lewis called upon D. E. French for a report of the Committee on Code for Testing and Rating Unit Heaters which had been working jointly with a Committee of the *Industrial Unit Heater Association* in the preparation of this Code.

Mr. French stated that the joint committee had prepared a Code for Testing and Rating Steam Unit Heaters and it had been presented to the *Industrial Unit Heater Association* on June 25 with the result that the association accepted the report of the joint Code Committee, with certain changes agreed to by the Committee, as an amendment to the rules for testing and rating which the association had previously adopted. The Association recommended that all member companies put the prescribed test procedure in use for trial and suggested that the present

committee be continued so that it might present the Code for adoption at the *Industrial Unit Heater Association's* meeting next January and revised to include any improvements that might result from the experience of members who use it.

Mr. French pointed out that in view of the action of the *Industrial Unit Heater Association* in adopting the report of the joint Code Committee as a tentative standard, the joint committee desired to report this action to the Society and felt that it would be ready to present a Standard Code to the members at the January 1930 meeting. For those interested Mr. French stated that copies of the Code with slight revisions approved by the Joint Committee would be available by application to the Society.

Chairman Lewis said that in view of the recommendations of the Committee to defer action in this matter, discussion on Professor O'Bannon's paper was in order.

Professor Kratz inquired about Professor O'Bannon's procedure in measuring the heat production of a unit with a thermocouple on a pitot tube and then explained the method that he had used to obtain satisfactory correlation of results. The speaker also asked whether Professor O'Bannon had had any difficulty in maintaining pressure sufficiently close to atmospheric so that air flow was not affected.

D. E. French, Philadelphia, was of the opinion that no basis existed for the assumption that there should be disagreement between results by measurement of air quantity by the pitot tube method and that of condensation temperature rise. He felt that the results obtained by Professor O'Bannon were a tribute to the methods used and the accuracy of the observations.

H. S. Wheller, asked whether conclusions had been reached in reference to obtaining correct results by any other method than that used.

S. R. Lewis, Chicago, said that a recent observation of the operation of unit heaters in a garage convinced him that the Code should provide some method of conducting proper tests on the job.

L. A. Harding, Buffalo, stated that there should be no difference between the two test methods if it is assumed that the steam tables are correct and the specific heat of air used is accurate.

W. A. Rowe, Detroit, believed that the difference of opinion between men who had studied this subject might be accounted for by the fact that some were in the fan business, accustomed to testing air and others were in the heater business, and were more accustomed to judging results on the basis of condensation of steam, with the result that one group preferred a method of measuring air direct and the other preferred to calculate air capacity by measuring temperature rise. Both were agreed as to the proper method of getting the B.t.u. rating from the condensation test, the main difficulty being that there is a prevailing opinion among engineers that the temperature rise method should give a greater air volume than the pitot tube method. For that reason the research work at the University of Kentucky was suggested in order to establish definite facts in this matter and the result indicates quite clearly that it is perfectly safe for all practical purposes to use whichever method is more convenient. Mr. Rowe thought that definite action on the adoption of the Code might be delayed because the results obtained in all investigations, while consistent, still left some question about the similarity of performance of a heater when disconnected from the chamber and operated on the floor.

Professor O'Bannon stated that Professor Kratz' comments hit the heart of the investigation and he pointed out that the slight percentage variations recorded represent experimental errors. According to Professor O'Bannon the main reason for the accuracy and consistency of the results was the thorough insulation of the mixing chamber where temperature readings were taken and for that reason numerous chamber divisions across the duct as suggested by Professor Kratz were unnecessary. With reference to other methods of testing no method has been proposed which can be used as a general scheme for all types of heaters. Professor O'Bannon stated that if final temperature is not important it is only necessary to erect a duct for measuring air velocities and to provide some method of insuring free delivery of the heater. The question of whether the mixing chamber reproduces natural delivery conditions is one that needs investigation.

D. E. French, Philadelphia, stated that field testing of unit heaters had not been given immediate consideration by the Code Committee as it was felt that the most important step was first to establish accurate laboratory methods by which manufacturers could test and rate unit heaters. The matter of field testing, however, is before the committee and will be its next job.

Second Session—Thursday, June 27, 9:30 a. m.

The meeting was opened by President Lewis. J. R. McColl, Detroit, stated that the news of the recent death of E. S. Hallett, St. Louis came as a distinct shock to everyone who knew him and that resolutions had been prepared and adopted by the Council which he thought should be presented to the meeting. The following resolutions were unanimously passed and an engrossed copy was to be sent to Mr. Hallett's family.

WHEREAS, Edwin S. Hallett, an Active Member of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, passed out of this life on March 29, 1929 and

WHEREAS, he was one of the Society's most loyal members, constant in attendance, contributing to the interest and value of the meetings by presenting his original ideas, and

WHEREAS, being widely known as an outstanding engineer in school house heating and ventilation practice, and

WHEREAS, the Society will miss his genial personality, his words of cheer and encouragement, his personal interest in each member and his kindly expressions of friendship, therefore, be it

RESOLVED, that we, the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, deeply regret the passing of our fellow member, Edwin S. Hallett, that we cherish the memory of his frank, upright character and that we hold in loving admiration his qualities as an independent and courageous thinker, and also, be it

RESOLVED, that a copy of these resolutions be spread upon the records of the Society and that a copy be sent to his family.

The first technical paper, Errors in Measurement of the Temperature of Flue Gases by Percy Nicholls and W. E. Rice, was read in abstract by A. V.

Hutchinson, Secretary. (Full text of paper to be printed in September 1929 JOURNAL.)

R. V. Frost, Norristown, said that much had been heard about the difficulty of accurately measuring stack temperatures and pointed out that the results of the authors' work indicated that very close accuracy was possible, particularly when the greatest degree of error shown was about 5 per cent.

A paper, Pipe Sizes for Hot Water Heating Systems by F. E. Giesecke and E. G. Smith (published in the June 1929 JOURNAL, p. 139) representing the result of co-operative research between the Society and the Texas Engineering Experiment Station was presented in abstract by Professor Giesecke.

H. H. Angus, Toronto, said that it was the common practice in Canada to use hot water systems in residences and public buildings, with circulating pumps and stated that with oil fired units the pump and the oil burner were connected to work simultaneously in order to produce satisfactory and efficient circulation.

Homer Linn, Chicago, asked Professor Giesecke whether he had determined the proper distance between the return connections to the main in proportion to the size of the radiator.

H. M. Nobis, Cleveland, noticed that bottom flow connections were used and stated that more effective heating might be obtained by following steam heating practice and making connections at the top.

R. C. Bolsinger, Philadelphia, asked whether there was any difficulty in the use of partitioned fittings as compared with regular fittings, with one pipe systems.

Professor Giesecke explained that the use of a pump to increase pressure head was desirable and it was a matter for the engineer to decide whether it is more economical to pay for the pump and its maintenance or to use larger pipe sizes and gravity circulation.

It was the author's opinion that the distance between the return connections could not affect a gravity circulating system. Relative to upper and lower supply connections for radiators, the speaker mentioned a series of tests conducted at the University of Illinois, where various combinations had been used, and it was indicated that the desirability of one or the other was dependent upon the type of radiator used.

In the present investigation changes in connection affected the pressure produced by the radiator riser as compared with the pressure head. The importance of using regular or partitioned fittings was unknown to the author.

In summing up, Professor Giesecke gave an analysis of the results of his investigations and said that a table of pipe sizes had been prepared for a direct return system now being built to represent a three-story installation having every possible piping arrangement, in order to find out whether the results are in accordance with the values determined and the assumptions that have been made.

President Lewis complimented Professor Giesecke upon his splendid presentation which brought out the fundamentals of the problem and made an apparently complicated subject very easy to understand.

S. R. Lewis, chairman of the Guide Publication Committee, was called upon to present his report:

THE GUIDE in the past has been generally divided into two sections—the Engineering Data Section, and the Catalog Data Section. The line of demarcation between these two sections has been rather sharply defined. It is the ambition of the Guide Publication Committee to make the line of demarcation less sharp. It is not desired to create the impression that certain engineering data which are valuable to engineers, are merely compiled and published with enough advertising from manufacturers to pay for printing the engineering data. There is nothing about the service which THE GUIDE performs, savoring of the advertising-studded specification covers of certain architects. There is nothing about THE GUIDE which demands advertising like that which may be found in certain college class books often: "With the compliments of a friend."

THE GUIDE, very definitely, is an instrument of service to both the advertisers and the engineers and the architects who consult its pages. It is desired that the Engineering Data Section and the Catalog Data Section shall co-ordinate with and supplement each other, and that THE GUIDE shall serve those of us who make things as well as those of us who specify that these things shall be used in buildings, and how they shall be used. To that end we are endeavoring to refer frequently in the Engineering Section to the Catalog Section, and to encourage contributors to the Catalog Section to refer to the Engineering Section. We are asking the advertisers to prepare their copy more as though it were going into the Engineering Section rather than as though it were going onto a billboard.

THE GUIDE 1929 is now used as an authority in thousands of offices, and its users, having looked up the technical data about any subject and wishing to specify the equipment by number, capacity or characteristics for some service, should find the information ready at hand in the Catalog Section of THE GUIDE.

We are developing more information about specifications with the hope that eventually suggestions to be published in THE GUIDE may standardize certain parts, and may make better business for the manufacturers and the contractors, somewhat in the same manner as the standard provisions of the contract of the *American Institute of Architects.*

Very fine support and co-operation from our engineering members has been given for THE GUIDE 1930. There will be 34 feature chapters. Some of those of THE GUIDE of 1929 have been omitted, and some additional subjects have been added. The nebulous separation between the heating section and the ventilating section has been eliminated entirely. The chapters have been re-grouped, starting with codes, standards and symbols; the discussion of heat losses; heating and cooling with air; heating with coal; heating with gas; heating with oil; heating with electricity. Then follow the various arrangements to carry and move the air and to transmit the heat from the various fuels to the rooms and to control it.

The chapters are all being rewritten in the light of these suggestions and arrangements. Of the 34 feature chapters, 23 are ready for the printer, except as to cross-index, and the missing 11 are all partly completed. We ask that the contributors to the Catalog Section shall always read the engineering data section before preparing their copy. We pledge our efforts to blend the two parts of THE GUIDE, and to make both parts as interesting and as valuable as possible.

In calling for discussion, President Lewis said that on his recent visit to St. Louis, where he represented the Society at the convention of the *Heating and Piping Contractors National Association,* he was delighted

to see the interest of that organization in THE GUIDE as informally expressed by the chairman of the Standardization Committee. He expressed the belief that the contractors organization and Society were working more closely together than in the past and felt that the two organizations were properly following out their functions; one in the scientific field and the other in the practical and commercial field.

Perry West, Newark, N. J., thought that the members should express their appreciation and assist Mr. Lewis in every way in his effort to produce THE GUIDE 1930.

W. H. Driscoll, paid a tribute to the work Mr. West had done in bringing the book up to a high state of perfection and said that he turned it over to a most worthy successor.

F. C. Houghten, director of the Research Laboratory, presented the results of recent investigations in a paper entitled, Capacity of Radiator Supply Branches for One and Two-Pipe Systems by F. C. Houghten, M. E. O'Connell and Carl Gutberlet (published in the May 1929 JOURNAL, p. 55).

When President Lewis called for discussion, W. H. Carrier, Newark, N. J., stated that a year or more ago when THE GUIDE was being compiled there were much conflicting data on return main sizes and accepted practice varied from three or more to one. For that reason, the Guide Publication Committee requested some research on the subject. He pointed out that an entirely different condition exists in continuous operation than in intermittent operation and in the latter case if pipe sizes are too small or if excessive vacuum is not eliminated, water hammer, water logging and other critical conditions occur. As a result of further research investigations it may be found that some other step than providing a large pipe size would be necessary in the practical solution of the problem.

Homer Linn, Chicago, inquired whether the condensing power of a radiator did not influence conditions in the branch connection and also asked if the difference in values in certain tests might not result from the air temperature surrounding the radiator.

Professor Giesecke expressed the belief that additional work should be done on two-pipe systems and suggested that a two-pipe system be defined as one in which the steam and condensate flow in the same direction. He also felt that further experiments should be made with higher steam pressures.

H. M. Nobis, Cleveland, said that his experience indicated that air elimination rather than the size of pipe was an important factor in the successful operation of a heating system. He also asked that return pipe sizes be calculated for gravity flow air for conditions known as atmospheric.

R. V. Frost, Norristown, proposed that the work on flow of air and water in return lines started by Professor O'Bannon be continued.

Perry West, Newark, N. J., expressed the opinion that one of the principle factors desired by the Guide Publication Committee was to determine the relation of the length of run both horizontally and vertically on the return pipe size.

L. A. Harding, Buffalo, stated that it was the intention of the Research Committee to continue the dry pipe tests started by Professor O'Bannon and said that there was need for work on other sizes, than 1 in. pipe, which had been completed. He said that the real need for information resulted from the use of the light weight and fin types of radiators, particularly on hot blast systems under thermostatic control. As yet no way has been found for correcting difficulties by changes in pipe sizes.

A written discussion contributed by Louis Ebin, Chicago, was presented by Mr. Houghten. The attention of the author was called to the fact that the laboratory's results seemed to be based on a definite point of critical velocity for every size of pipe for both one and two pipe systems and a diagram accompanying the discussion showed a series of curves for 1 in. pipes of various lengths in both horizontal and vertical positions, and indicated a condition not brought out in the laboratory report as it shows that the critical velocity varies definitely with the length of the pipe. It was noted that sufficiently covering the riser of a two-pipe system might increase its capacity and engineers might find as a result of further experimental work, that steam capacities of pipes are far in excess of anything used or advocated at present and that in the future results may be obtained with steam heating systems using smaller pipe sizes than present day engineers are accustomed.

In answering questions, Mr. Houghten stated that when an increased quantity of steam goes through a two-pipe riser, the capacity pressure drop relationship will reach a point where the capacity of the pipe will not increase even with a considerably higher pressure, and if the pressure increase be several times that necessary to get maximum capacity, condensation will be forced into the riser or through a pitched pipe into the radiator. It was explained to Mr. Linn that with a considerable variation in the quantity of steam carried in the riser, no variation was apparent in the capacity of the branch.

President Lewis announced the appointment of a Committee on Resolutions with the following personnel: W. H. Driscoll, Chairman, H. P. Gant and C. L. Riley.

Report of Endowment Fund Committee

The Secretary, A. V. Hutchinson, read the report of the Special Council Committee relating to a proposed Society Endowment Fund which had been prepared and sent out to all Society members prior to the Summer Meeting.

May 24th, 1929.

TO THE MEMBERS
OF THE AMERICAN SOCIETY OF
HEATING AND VENTILATING ENGINEERS:

At the Semi-Annual Meeting of the Society, June 26, at Bigwin Inn, Lake of Bays, Ontario, Canada, the report of a Special Committee of the Council, relating to the creation of a Society Endowment Fund, will be presented for the approval of the A.S.H.V.E. members.

The Committee, A. C. Willard, Chairman, John Howatt and W. T. Jones, was appointed by resolution of the Council March 8, 1929, and the Council received the report of the

Committee and approved it at the May 13, 1929 Meeting.

The report of the Committee and the resolutions of the Council follow:

THE SOCIETY ENDOWMENT FUND

(1) In order to conserve certain specified profits accruing to the Society from certain of its agreements and activities, it is proposed to establish a Society Endowment Fund* to be administered solely by the Council as a Depository for the net profits from the agreement with Domestic Engineering Publications dated January 10, 1929, and such other balances, receipts, donations and bequests as may be received from time to time and specifically designated by the Council of the Society to be paid into the Society Endowment Fund.

(2) This Fund is to remain for all time a permanent endowment fund, and only the income from the Fund may be devoted to such activities, purposes or uses of the Society as the Council may approve by a favorable formal vote of not less than three-quarters of the entire Council membership.

(3) The income from the Fund remitting from the agreement with Domestic Engineering Publications is to be restricted entirely to the research activities of the Society and can be applied to no other purpose except by formal action of the Council in the manner already indicated in paragraph (2).

(4) This proposal is to be sent by mail before June 1, 1929, to the entire membership of the Society as a recommendation of the Council, and is to be voted on at the June 1929 meeting of the Society.

April 30, 1929.

Submitted by the
COMMITTEE ON ENDOWMENT FUND
John Howatt,
W. T. Jones,
A. C. Willard, *Chairman.*

On motion of Mr. Langenberg, seconded by Mr. Carrier, it was

VOTED: That the report of the special committee be accepted and that the Trustees to administer the fund be the President of the Society, the Treasurer and the Chairman of the Finance Committee.

and it was further

VOTED: That this report be placed before the Semi-Annual Meeting, 1929, with the Council recommendation of approval.

THORNTON LEWIS, President. A. V. HUTCHINSON, Secretary.

* This new fund is not to be confused in any way with the present Research Laboratory Endowment Fund, which is entirely separate and distinct from the new Society Endowment Fund.

President Lewis stated that the report was before the meeting for discussion and called attention to the action taken by the Council.

Summarizing the points which had been considered by the Council, President Lewis stated that it was felt unwise to designate the income for use entirely for Research Laboratory activities and for that reason, it was thought desirable that an Endowment Fund be set-up leaving it to future Councils to appropriate the income for any purpose which might be desirable. While it is the Council's function to appropriate moneys belonging to the Society for any purpose, the members felt it advisable to get the advice of the Society to find out the wishes of the members. The Council will obtain legal advice when it formally acts in this matter.

H. P. Gant, expressed the opinion that it was a far sighted movement and believed that the Endowment Fund in times of stress, which had previously been experienced by the Society, would be of tremendous value.

F. W. Herendeen stated that he was not in agreement with the idea of tying up the principal but felt that the establishment of a fund to be governed and spent by the Council, would be a good thing. Therefore, he felt, it would be wise for the members to consider the restriction stated in the resolution permitting the spending of only the income.

N. B. Hubbard, Detroit, suggested that a clause be included authorizing the Council by a 100 per cent vote, to spend the principal in case of emergency.

Homer Linn, Chicago, was of the opinion that bequests received with the understanding that they would go into the fund "for all time," could not be spent on the basis of a vote by the Society or the Council.

J. A. Donnelly, West Virginia, expressed the thought that it would be better to have the fund available for use with proper safeguards rather than have it tied up permanently. He suggested that a reasonable reserve fund be established and let the Council spend such money as comes to the Society as it may deem best.

A member of the Committee, John Howatt, pointed out that the intent of the resolution was clear namely, that the income from the fund was to be used for research and that the moneys received by the Society be safeguarded so that the income be usable in a practical and satisfactory manner no matter what the development in the field of heating and ventilating engineering might be.

W. T. Jones, Boston, another member of the Special Committee expressed himself as in favor of the idea of a permanent fund, the interest to be used as the Council may direct. It has occurred to the Council and others that moneys accumulating can immediately be used to reduce dues but that would be a backward step, he thought. The starting of an endowment fund suggested as has been done by other organizations is a forward step and will be of great value to the Society at some future time. As a member of the Committee, he said that it would be interesting to have the point settled in reference to the permanency of the fund.

W. H. Driscoll said that the resolution was laudable in its purpose but it would be questionable for the Society to go on record in favor of the resolution as drawn or on a substitute resolution with any idea of summary action in disposing of the matter. The legal aspects of the problem should be seriously considered. He suggested that the matter of establishing an endowment fund be referred back to the Council so that if in its wisdom and judgment the Council determines that it is advisable to establish a fund and have definite recommendations available for members prior to the Philadelphia Meeting and take action in January.

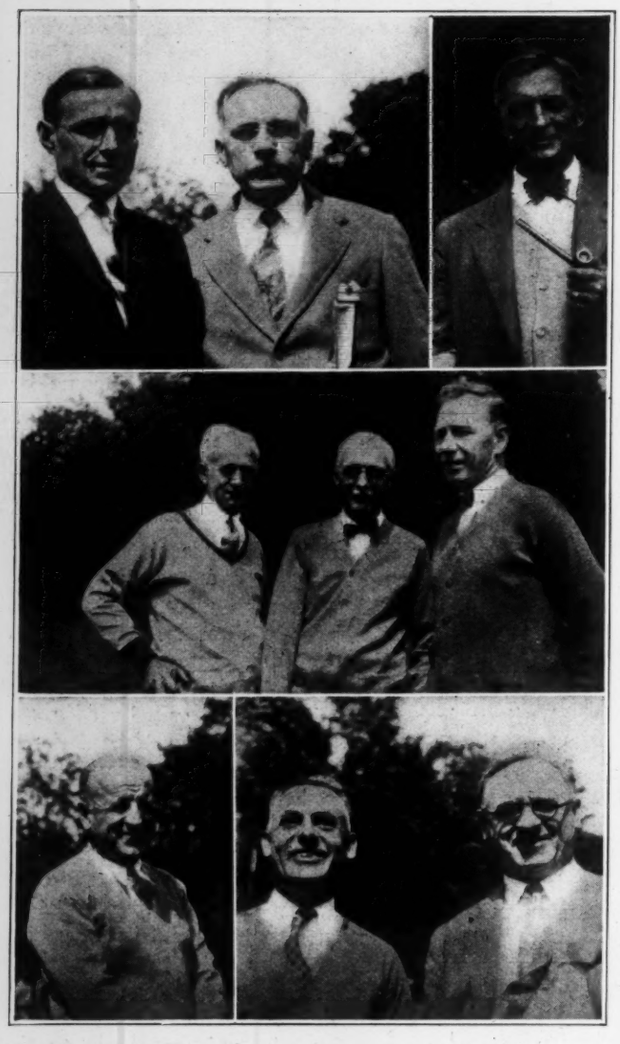
This was made as a motion and seconded by John Cassell and unanimously passed.

L. A. Harding, Buffalo, reported as Chairman of the Continuing Committee on Codes for Testing and Rating Steam Heating Solid Fuel Boilers and said that three definite items would be brought before the meeting.

First the Committee requested approval of some slight changes in the A.S.H.V.E. Standard and Short Form Heat Balance Code for Testing Low Pressure Steam Heating Boilers, designated as Codes 1 and 2. The changes recommended by the Commit-

tee were the definitions of the term "grate area" and of the factor of evaporation.

Mr. Harding explained that, in view of the Society's expressed attitude relative to the rating of boilers on the basis of tests, it was necessary to have a satisfactory performance code and as the heat balanced codes are not well adapted for rating tests, a little different form designated as A.S.H.V.E. Performance Code for Testing Solid Fuel Boilers (Code No. 3) was submitted. This Code he said is identical except in one or two minor cases with the Standard Boiler Code recently adopted by the *National Boiler and Radiator Manufacturing Association.* The reason for the slight changes made by the Society is to put this Code in conformity with the values used in the Code of Minimum Requirements for the Heating and Ventilation of Buildings, where the latent heat of evaporation is given as 971.7 as compared with the value used by the manufacturers of 970.4.



TOP ROW: A. V. HUTCHINSON, SECRETARY; THORNTON LEWIS, PRESIDENT; W. H. CARRIER, SECOND VICE PRESIDENT. MIDDLE ROW: C. V. HAYNES, PHILADELPHIA; J. R. MCCOLL, DETROIT, AND JOHN HOWATT, CHICAGO. BOTTOM ROW: W. H. DRISCOLL, NEW YORK; PERRY WEST, NEWARK; PROF. G. L. LARSON, MADISON, WIS.

On motion of R. V. Frost, properly seconded, the revisions of Codes 1 and 2 were unanimously adopted.

Motion for the adoption of Code 3 was made and seconded and unanimously carried.

Mr. Harding stated that the test codes were rather easy to handle, but the matter of the rating code was quite a different story and submitted a details report of the work of his Committee as follows:

Report of A.S.H.&V.E. Continuing Committee Codes for Testing and Rating Steam Heating Solid Fuel Boilers

THIS committee met in Buffalo May 4, 1929, to tabulate and discuss the replies received to the questionnaire previously mailed to a list of boiler manufacturers and others interested in this matter. This questionnaire was printed in the April, 1929 issue of the JOURNAL of the A. S. H. V. E. Replies were received to the questionnaire from a large percentage of the membership of the *National Boiler and Radiator Manufacturers' Association,* some independent manufacturers of steel and cast iron heating boilers; also the following reply from the Boiler Output Committee of the *Heating and Piping Contractors National Association*:

"Your letter of February 11, 1929, having reference to code adopted by the A. S. H. V. E., at their annual meeting in January this year and having further reference to questionnaire prepared by a continuing committee charged with the duty of first determining the feasibility of applying this code in practice and looking for criticisms, opinions and suggestions from other interested associations.

"Referring directly to the questionnaire prepared and replying to No. 1 will say—Yes, but the condition of operation should be standardized. In addition the manufacturer should publish the following—kind of fuel and B.t.u. value, rate of combustion, firing period, attention period, flue gas temperature, stack draft, over-all efficiency, dimension of chimney and available fuel charge.

"In answering the first paragraph of question No. 2, we are in favor of a No. 1 rating with a complete performance chart.

"The second paragraph of question No. 2 has our same answer, prefer complete performance chart.

"Third paragraph of question No. 2—our answer is 'No.'

"Answering the 4th paragraph of question No. 2—published rating tables should have no bearing on determining loads so far as boiler manufacturers are concerned. Complete performance chart should be furnished.

"Fifth paragraph of question No. 2—our answer is complete performance chart to be furnished.

"Third question first paragraph—our answer is 'No.'

"Second paragraph of question No. 3—our answer is, that manufacturer should not employ any percentage in his published rating tables.

"This same applies to paragraphs No. 3 and No. 4 of question No. 3.

"Answering question No. 4—we prefer to see Boiler Output values stated in direct radiation based on 225 B.t.u. for steam and 150 B.t.u. for water radiation.

"In answering question No. 5 including first and second paragraphs, our answer is, that complete performance charts should be furnished, also actual grate dimension when measured by the *Heating and Piping Contractors National Association's* standard method, also outside dimensions of boiler, height of water line and depth of ash pit.

"In answering question No. 6, our answer is, that all round boilers should be tested. Rectangular boilers, a suffi-

cient number in each series, should be tested so that a reasonably accurate performance chart can be determined for the complete series of boilers.

"We regret that the answers to your questions did not receive earlier attention but it was entirely overlooked until a wire was received from our main office in New York.

Yours very truly,

HEATING AND PIPING CONTRACTORS NATIONAL ASSOCIATION

BOILER OUTFIT COMMITTEE

(*Signed*) Geo. M. Getschow, *Chairman.*"

The members of the heating division of the *American Boiler Manufacturers Association* did not, except in one instance, favor the committee with any replies to the questionnaire.

A summary of the replies received follows:

1. All are agreed that manufacturer should guarantee the listed ratings of the boilers manufactured under conditions of operation he may state and dimensions of chimneys given by him.

2. Several manufacturers apparently, favor a one number rating, one of which states that "it would be advisable to have a one number rating for price comparisons of boiler capacities, other capacities with performance characteristics are desirable."

Many of the answers given to this question are in effect, "we do not favor 'one number rating' with this exception, that, considering conditions in the industry, and this is entirely a commercial reason, the one number rating would go to some length to insure uniform price."

3. The great majority of the replies received indicate that a "multi-number rating" is favored.

The Heating and Piping Contractors National Association express a preference for complete performance charts.

4. Practically all prefer to state boiler outputs in both B.t.u. per hour and sq. ft. of equivalent direct radiation.

5. The consensus of opinion in reference to the statement of conditions that should accompany rating tables are practically uniform on the following items:

(1) Combustion rate
(2) Firing period, attention or fuel available in hours
(3) Overall efficiency
(4) Draft tension
(5) Dimensions of chimney
(6) Fuel (kind not stated) 12,500 B.t.u. per lb.
(7) Grate area and chimney dimensions to be given
(8) The majority of the answers omit flue gas temperature as a condition. One suggests a limit of 800 deg. fahr.

We believe that the thoughts of this committee and two associations previously mentioned, are apparently in harmony covering the principal features of a boiler rating code.

If a prospective purchaser is able to obtain from the literature of the manufacturer in advance of the actual purchase, guaranteed definite information of what he may expect from the operation of the apparatus as to *output, economy* and *attention* under a specific set of conditions and over a reasonable range of operation, he will be provided with the necessary information required to select the size and apparatus to suit his particular condition and at the same time be able to predict the operating economy with reasonable accuracy which he may later expect to obtain.

This information can only be obtained through the medium of standardized boiler tests or trials.

It is obvious that no real progress could be expected in formulating a boiler rating code until a satisfactory boiler performance testing code be devised to give the information sought. Such a code is now in satisfactory operation by a number of the boiler manufacturers.

We believe the preceding paragraphs cover the principle involved on the subject of boiler rating.

The committee, after its meeting on May 4, formulated a tentative revision of the present A. S. H. V. E. Code for Rating Low Pressure Heating Boilers, which was mailed to the same manufacturers and organizations that received the questionnaire. The proposed revision was accompanied by a copy of the committee's comments on the subject of boiler rating which follows:

GENERAL

A rating code for any type or kind of apparatus must of necessity include the following items:

1st—The output.
2nd—Specified conditions of operation for the output stated.
3rd—The limits placed on the specified conditions.

It is obviously necessary to standardize limits for some of the specified conditions, otherwise a rating code could not fulfill its function.

Some of the limits to conditions are naturally now set by custom and usage or by ordinances or laws, designed to protect the health or safety of the community.

For example: Custom and usage have decreed that the conditions relative to steam pressure for rating purposes shall be 2 lb. gage at the boiler for heating boilers, whereas, overall efficiency, draft tension, temperature of flue gas and rate of combustion are conditions over which it is difficult if not altogether impractical to assign definite limits in the present state of this art with the possible exception of flue gas temperature. This should naturally be limited to a maximum value consistent with safety relative to fire hazard.

Placing a minimum value limit on boiler efficiency covering the average load period of the heating season, would appear to be in line with a program of fuel conservation and probably receive a welcome by prospective owners of heating boilers.

Not the least item, however, in boiler economy is the manner in which the boiler is actually operated by the owner and over which the manufacturer has no control.

The manufacturer can and does produce boilers which give relative high efficiency when properly operated but who can set a minimum efficiency limit that would produce the actual results that are apparently desirable.

There are a number of terms employed in this art, the meaning of which is either vague, not clear or by no means standardized. We give below the definition of various terms as we understand them.

Purchaser:

Construed to mean the person responsible for the selection of the boiler.

Equivalent Direct Radiation:

The heat emission of 240 B. t. u. per hour per square foot of manufacturers *rated surface* of direct steam radiation and 150 B.t.u. per hour per square foot of manufacturers *rated surface* of direct hot water radiation.

One Number Rating:

A single rating stated for each boiler listed in a manufacturer's catalogue.

Multi-Number Rating:

Two or more ratings stated for each boiler listed in a manufacturer's catalogue.

Heating Boiler Output:

As defined by the proposed A. S. H. V. E. Performance Test Code for Steam Heating Solid Fuel Boilers.

Boiler Efficiency:

The overall efficiency of grate and boiler as defined by the proposed A. S. H. V. E. Performance Test Code for Steam Heating Solid Fuel Boilers.

Estimated Maximum Load, Peak Load, Starting-Up Load:

These terms are considered synonymous and construed to mean the load, stated either in B. t. u. per hour or equivalent

direct radiation that has been determined by the purchaser to be the greatest estimated output that the boiler will be called upon to carry in operation. We recommend the use of the term estimated maximum load in this connection.

Net Load:

This term now has various meanings, all the way from the load of the radiation in normal operation to the maximum load. We recommend that this term be dropped from heating literature as serving no useful purpose.

Estimated Design Load:

The load, stated in B. t. u. per hour or equivalent direct radiation, as estimated by the purchaser for the conditions of inside and outside temperature for which the amount of installed radiation was determined and is the sum of the heat emission of the radiation to be actually installed plus the allowance for the heat loss of the connecting piping plus the heat requirements for any apparatus connected with the system.

Estimated Average Load:

The estimated average load stated in B. t. u. per hour or equivalent direct radiation for the heating season and based on the average outside temperature during the heating season for the locality in question.

Heating-Up Factor:

The percentage added to the estimated design load in determining the estimated maximum load.

Attention, Firing Period, Fuel Available in Hours:

The hours required to burn one available fuel charge. *Fuel available* is defined in the proposed performance boiler test code. We recommend the discontinuance of the use of the terms *attention* and *firing period*.

Load Rating Versus Output Rating:

These terms are not synonymous as now employed by various manufacturers. Some manufacturers prefer to include in their rating the allowance for the heat loss of connecting piping, or the heating-up factor or both, in either case the ratings stated are obviously not the boiler outputs required for either the estimated design load or the estimated maximum load, but are evidently some lower figure.

It does not appear to the committee that the boiler manufacturer should assume the responsibility of guaranteeing a rating based on the use of some heating-up factor of which the purchaser has no definite knowledge.

We believe the responsibility for the selection and application of a heating-up factor lies entirely with the purchaser just as much as it is his responsibility for determining the estimated design load. If he is capable of the latter he surely should be and is naturally capable of the former.

If a manufacturer includes in his literature the necessary heat transmission data for calculating the estimated design load he might very properly include a table of heating-up factors.

He does not guarantee the former so why should he guarantee the latter.

We do not see that the necessary process of selecting a boiler for a given load is in any way simplified by including in the published rating designation allowances for various items of estimated value.

Boiler outputs are something quite tangible when determined by a uniform test code, whereas, boiler loads are only estimates.

If all ratings stated in catalogues were boiler outputs as determined by the provisions of the test code, the boilers of various manufacturers would evidently be more easily compared, which overcomes one of the greatest objections frequently referred to as the existing confusion in boiler ratings.

In the interest of uniformity in catalogue listing, this committee recommends that all ratings be stated in terms of boiler outputs. The terms, boiler load rating and boiler output rating, would be synonymous and we see no reason for employing the term boiler load rating.

Range and Number of Boiler Outputs Desirable per Boiler:

If the purchaser is to have any very definite idea of the economy of the boiler he proposes to install he should naturally be provided with the efficiency corresponding to the average output as determined by and equal to the estimated average load estimate. This is the lowest output that there would be any very good reason to list.

The estimated average load in localities where heating is important is frequently assumed to be approximately 35 per cent of the estimated design load.

If the heating-up factor is assumed as 65 per cent then the "estimated maximum load" becomes 165 per cent of the estimated design load and the estimated average load is 21.2 per cent of the "estimated maximum load."

It is recognized that these percentages are somewhat variable but it would appear reasonable to assume that the minimum output listed for a particular boiler should bear some relation to the maximum listed output and we suggest that approximately 20 to 25 per cent be employed in this connection. Thus, if 25 per cent was used the fuel available in hours for the minimum output would be 6.6 times that of the maximum output listed.

It is recommended that at least five outputs be given with the corresponding conditions for each boiler listed in order that those who are interested in the use of boiler performance charts may have the desired information to plot such charts. This would require listing three outputs fairly equally spaced between the minimum and maximum.

"Fuel Available in Hours" for Maximum Output:

If 8 hours be considered a desirable time as set by custom and usage for the smaller size boilers to employ for the design load, then on the basis of a 65 per cent heating-up factor the fuel available in hours for maximum output becomes 4.84 hours. If 6 hours is satisfactory for the larger boilers to employ for the design load the fuel available in hours for maximum output becomes 3.7 hours.

In the interest of uniformity in rating it would be well to place some limits on the fuel available in hours for the maximum outputs listed, although we have no recommendations to make at this time.

Flue Gas Temperature:

It is coming to be more generally recognized that flue gas temperature *is not* a reliable indicator of efficiency and in many cases only remotely connected with efficiency.

The information in reference to both the temperatures of flue gas and draft tension is, we believe necessary, particularly for boilers of the larger size.

The purchaser must have this information if he is to be able to perform the necessary calculations in determining satisfactory Chimney or stack dimensions.

This is more particularly true for boiler installations serving high buildings.

If only draft tension is stated the purchaser is left to make his own estimate as to stack temperature. The manufacturer should be as much concerned in the satisfactory operation of the installation as the purchaser and this is hardly possible unless a proper size chimney or stack is provided by the purchaser.

Boiler Test Codes:

It is proposed to recommend perhaps several minor changes in several items of the A. S. H. V. E. Code for Testing Low Pressure Steam Heating Solid Fuel Boilers.

We also propose to recommend to the Society the adoption

of the test code now employed by the *National Boiler and Radiator Manufacturers Association* and to be designated as the A. S. H. V. E. Performance Test Code for Steam Heating Solid Fuel Boilers.

This Committee met with representatives from the *American Boiler Manufacturers Association* and the *National Boiler and Radiator Manufacturers Association* in Buffalo, June 20, 1929 for the purpose of discussing the proposed revision of the present A. S. H. & V. E. Code for Rating Low-Pressure Heating Boilers.

As a result of this conference your Committee and the representatives from the *National Boiler and Radiator Manufacturers Association* agreed upon the following draft of a rating code. The representatives present at this meeting from the *American Boiler Manufacturers Association* did not agree with this code. Their views on the question of rating heating boilers are expressed by the letter which follows:

We, the undersigned manufacturers of low pressure steel heating boilers, representing approximately 95 per cent of the total steel heating boiler industry, object to the Code for rating low pressure heating boilers as proposed by your Committee, dated May 17, 1929. The objections are based on the following reasons:

1. The selection of boilers as based on the maximum capacity, would result in smaller steel boilers being used on a specific job as compared to present practice, with a consequent lowering of operating efficiency.

This Code specifies minimum requirements, and it has been experienced that in similar Codes, where minimum requirements are given, minimum requirements are used.

The present practice in selecting steel boilers gives a uniformly higher reserve capacity than as proposed by the Code.

2. The proposed Code does not consider heating surface, which is a prime essential in determining boiler capacity or ratings, and on which basis steel heating boilers have been rated and satisfactorily selected for a great number of years.

3. We object to the use of a multi-rating, as it would tend to cause confusion in an industry that has used but a single rating satisfactorily for a number of years.

For Testing and Rating Steam Heating Solid Fuel Boiler

4. We object to the use of anthracite fuel only as a basis for determining rating.

Our recommendations regarding the selection and rating of heating boilers are based on a practice which has proven to be satisfactory to both Manufacturer and User throughout the entire life of the steel heating boiler industry, and are as follows:

(1) The published boiler rating shall correspond to the "estimated design load" as defined in *Section* 5 of the Code of Minimum Requirements, being the sum of Items A, B and C in Paragraph 1.

(2) The published boiler rating shall be determined by the amount of heating surface and grate area.

We are prepared to furnish your Committee with exact information as to minimum ratios of rating to heating surface and heating surface to grate area to be employed in determining the rating for heating boilers.

Yours truly,

Kewanee Boiler Corp.	Titusville Iron Works Co.
Pacific Steel Boiler Corp.	Fitzgibbons Boiler Co., Inc.
Heggie-Simplex Boiler Co.	Ames Iron Works
Oil City Boiler Works	The Brownell Co.
International Boiler Works Co.	Erie City Iron Works

Your Committee feels that they have secured the support of a large percentage of the manufacturers of heating boilers and submits the following revision of the present boiler rating code for your consideration.

Proposed Revision of January, 1929 A.S.H.V.E. Code for Rating Steam Heating Solid Fuel Boilers

(1) PURPOSE

The purpose of this Code is to standardize the method to be employed and followed by any person, partnership, firm, corporation or association, who may desire to make use of, or employ for any purpose whatsoever the statement. "The rating of the boilers herein listed are in accordance with the provisions of the A. S. H. V. E. Code (year) for Rating Steam Heating Boilers Burning Solid Fuel."

(2) RATING DESIGNATION

It is understood that all ratings stated are boiler outputs for the corresponding boiler designation as were determined and defined by the provisions of the "A. S. H. V. E. Performance Test Code for Steam Heating Solid Fuel Boilers (year)" and as governed by the conditions as set forth under paragraph 4 and accompanying the ratings.

The output for each boiler shall be stated in thousands of B.t.u. per hour and also in sq. ft. of equivalent direct radiation. It shall be optional to state, in addition to the two methods indicated, the output in lb. of steam per hour.

(3) RANGE OF OUTPUTS FOR EACH BOILER LISTED

There shall be stated a minimum number of five boiler outputs for each boiler listed. The outputs shall have a range from maximum output to approximately 35 per cent of the maximum output and the intermediate outputs given are to be approximately equally spaced between the minimum and maximum outputs.

(4) LIST OF CONDITIONS, STATEMENTS OF LIMITING CONDITIONS AND MANUFACTURER'S GUARANTEE

There shall be stated under each output listed the numerical values for each of the following five items:

1. Fuel available in hours.
2. Combustion rate lb. per hr. per sq. ft. of grate surface.
3. Overall efficiency per cent.
4. Average draft tension, inches water.
5. Interior dimensions of chimney and height.

The following statements shall be included under each table of rating.

"The *priming* for any output listed above does not exceed two (2) per cent."

For Anthracite Fuel

"The ratings are based on a steam pressure of 2 lb. gage at the boiler and anthracite coal stove size, having a calorific value of 12,500 B. t. u. per lb. on a moisture free basis."

For Bituminous Fuel

"The ratings listed are based on a steam pressure of 2 lb. gage at the boiler and Bituminous Coal 3 in. by 2 in. size, having a calorific value of 13,000 B. t. u. per lb. sulphur content not exceeding 2 per cent and volatile content of not less than 30 per cent on a moisture free basis."

For Coke Fuel

"The ratings listed are based on a steam pressure of 2 lb. gage at the boiler and by-product or gas coke of commercial size best suited to the boiler."

"The inside dimensions and height of chimneys listed should be satisfactory when properly constructed and having no other opening except for the purpose of serving the boiler and when free from the effect of adverse air currents. Allowance should be made for any other chimney openings,

elbows in the smoke flue or breeching and for extra long smoke flue or breeching."

(5) TABLE OF DIMENSIONS

A comprehensive table of dimensions of the boilers listed shall be included in the same bulletin or catalogue with the ratings. This table shall include the number and pipe size of steam and return connections and location, smoke flue dimensions and height above floor line, grate area and height of boiler water line and such other dimensions as may be required for properly indicating the boiler to scale on a set of complete heating installation plans.

(6) DEFINITIONS

Purchaser:

Construed to mean the person responsible for the selection of the boiler.

Manufacturer:

The individual, firm or corporation who manufactures the boilers for which corresponding ratings are listed.

Boiler Output:

As defined by the proposed A. S. H. V. E. Performance Test Code for Steam Heating Solid Fuel Boilers.

Estimated Maximum Load:

Construed to mean the load stated in B. t. u. per hour or equivalent direct radiation that has been estimated by the purchaser to be the greatest or maximum load that the boiler will be called upon to carry.

Equivalent Direct Radiation:

Construed to mean the heat emission of 240 B. t. u. per sq. ft. of manufacturer's rated surface of direct steam radiation and 150 B. t. u. per hour per sq. ft. of manufacturer's rated surface of direct hot water radiation.

Grate Area:

As defined by the A. S. H. V. E. Code for Testing Low Pressure Steam Heating Solid Fuel Boilers.

Fuel Available in Hours:

Construed to mean the hours required to burn one available fuel charge. The available fuel is defined by the proposed A. S. H. V. E. Performance Test Code for Steam Heating Solid Fuel Boilers.

Overall Efficiency:

As defined by the proposed A. S. H. V. E. Performance Test Code for Steam Heating Solid Fuel Boilers.

Priming:

The amount of free moisture carried by the dry saturated steam vapor delivered by the boiler stated as a percentage of the total weight of the sum of the dry saturated steam plus the free moisture delivered.

Signed:

L. A. Harding, *Chairman*

R. V. Frost F. C. Houghten

A. S. H. V. E. Continuing Committee Codes for Testing and Rating Steam Heating Solid Fuel Boilers

In summarizing the results of the Committee's work, Mr. Harding stated that the proposed revision of the January 1929 A. S. H. V. E. Code for Rating Steam Heating Solid Fuel Boilers simply requires a complete picture of boiler performance on the basis of output rating and from the data derived, the purchaser can draw his own performance curves, calculate his loss from the structure and apply his own heating-up factor. The Committee does not bring this report for approval or adoption but merely for discussion, he said.

Professor Kratz, Urbana, felt that a complete picture of boiler performance could not be obtained without including flue gas temperature.

J. F. McIntire, Detroit, suggested that the name of the Code be changed from rating to output code and added that a report of the Committee be received and that further recommendations be made at the Annual Meeting in January. The motion was seconded and unanimously carried.

Third Session—Friday, June 28, 9:30 a. m.

The third session opened with the presentation of the paper entitled Determining the Quantity of Dust in Air by Impingement by F. B. Rowley and John Beal, which was published in the July 1929 JOURNAL, page 233. Professor Rowley presented the paper in abstract and P. D. Close read several written discussions.

Margaret Ingels, New York, expressed the opinion that the dust determining instrument should take into account all of the dust in the air sampled, that is, all of the particles, fumes and smoke it is possible to remove with the best of present materials and knowledge at hand. A written discussion by Kimball and Hand presented comparative data on dust counts by the Hill and Owens instruments, and stated that a greater degree of accuracy was apparently possible with the more costly Owens device and that inasmuch as it appears that accuracy has been sacrificed in the construction of the Hill Dust counter for cheapness of construction, experienced hands were necessary in the use of the latter device to obtain satisfactory results.

E. C. Evans, Pittsburgh, asked if there was any limiting size mesh or size of dirt which would permit a definite statement of filter efficiency. Homer Linn, Chicago, inquired about the effect of the volume of the test chambers on the dust count. In answer to these questions, Professor Rowley stated that investigations made so far indicate that there is a practical limit at which the dust count will give relative values and that standards have been established for definite classes of filter work.

The next speaker was P. D. Close, who presented his paper entitled Five Suggested Methods of Appraising Insulations, published in the July 1929 JOURNAL, p. 245.

Written discussions by H. J. Schweim, Chicago, H. B. Lindsay, Chicago and P. D. Codwise, Buffalo, were submitted.

Mr. Schweim expressed the opinion that a method of comparing insulations is needed, but contended that such a method should be based on overall transmissions through the completed wall. In speaking for Mr. Lindsay, G. T. Pearce, Chicago, pointed out that in undertaking a classification of insulating materials, we are contending with a number of varying conditions, and that the method of proceeding is by careful and competent tests under actual conditions of use, as far as that is practicable.

Mr. Lindsay was favorably impressed with the idea of a method of appraising insulations, and was particularly concerned over the matter of the many unwarranted insulation claims being made. The contribution of P. W. Codwise concurred with the opinion expressed in the paper that conductivity figures alone do not tell the entire story about the

efficiency of insulating materials. He pointed out that greater emphasis should be given to overall transmission figures.

J. H. Bracken, Chicago, contended that the difficulties described in the paper are sales difficulties, and argued for the sufficiency of the hot plate method of testing as a basis for solving the problems involved. Mr. Bracken opposed the paper on the ground that it tends to aggravate the malady it aims to cure, and lamented the fact that various materials had been given different classifications, even if only to show the principle involved, because such classifications, be believed, will be immediately construed by many people as standards, and may result in a certain amount of injustice to some manufacturers.

J. D. Cassell, speaking as a purchaser of insulation, did not think it was the function of the Society to go into the testing of materials in a building.

Prof. F. B. Rowley, Minneapolis, pointed out that the confusion now existing with respect to rating insulations is due largely to the fact that the heat transfer through materials is given on the basis of an inch of thickness, whereas they are sold or installed in different thicknesses. Professor Rowley suggested the possibility of rating insulations according to the conductance or resistance of the thickness installed or manufactured similar to the second suggested method of the paper.

Homer Linn, Chicago, expressed thanks to the author of the paper for starting this discussion, even though no definite conclusions are reached. He took exception to the claims of certain manufacturers concerning the insulating qualities of their materials, when such claims do not seem to be well founded.

The opinion was expressed by Prof. A. P. Kratz, Urbana, Illinois, that the whole matter is more or less one of sales psychology, and that the engineer is really not actually concerned with the rating of insulations since, if he has the fundamental constants, he can make his calculations and base his conclusions on whatever the results of his calculations may be.

J. G. Shodron, Ft. Atkinson, Wisconsin, brought up the question of infiltration through various insulating materials, and wanted to know how important this item is so far as the rating of materials is concerned. In answer to this question, Mr. Close stated that although the infiltration through some materials was a great deal more than through others, due to the difference in porosity, this item was reduced to a negligible quantity when the material is installed in a wall with other materials, particularly when the wall is plastered. In this connection Mr. Pearce pointed out that it is the practice of his organization in refrigeration work to insist upon air-tight and moisture-proof construction in order to obtain the maximum possible efficiency of the insulation.

L. A. Harding, Buffalo, was of the opinion that it would not be practical to attempt to rate insulations by taking into consideration every conceivable factor which might enter into the problem, particularly if the labor element, which varies tremendously, is one of these factors. Mr. Harding was doubtful as to the necessity for a method of classifying insulations.

In reply to the discussions, Mr. Close stated that as had been anticipated, the subject apparently resolved itself into two divisions, those who are in favor of a method of rating or classifying insulations, and those who are opposed to the idea. He agreed with Messrs. Harding and Lewis that if the attempt were made to take every conceivable factor, particularly labor costs, into consideration, the problem becomes hopelessly complicated. It was stated that from the purely engineering standpoint, costs would not be a factor, but that costs would of necessity require consideration in the selection of materials.

In answer to Mr. Schweim's statement that the average person would not know what was meant by unit of resistance used in the fourth suggested method, it was pointed out that this argument applies equally to all of the phraseology used in connection with the technical phases of insulation. Referring to Mr. Bracken's analogy in which he claimed that insulations can no more be rated on a rational basis than can wearing apparel, Mr. Close pointed out that although there are certain commodities that are of such a character that they cannot be properly rated or classified, there are others which are of a sufficiently tangible nature that they can be readily rated, and in fact, this is now being done in many different ways in this Society and in other organizations.

Announcements were made by R. C. Bolsinger, Acting Chairman of the Nominating Committee, and E. C. Evans, Secretary of the Committee, concerning the recommendation of candidates for office in 1930.

The next paper presented was that entitled, Time Lag as a Factor in Heating Engineering Practice by James Govan after a few preliminary remarks. (Published in July JOURNAL, p. 239.) Mr. Arkley of Ontario, in discussing this paper, stated that in his opinion Mr. Govan's suggestion concerning the stating of resistance in terms of hours instead of degrees, was well taken. He also stated that it was his opinion it would be some time before buildings would be built with walls having a resistance factor of from eighteen to twenty hours.

S. R. Lewis endorsed the paper and expressed his appreciation for the constructive criticism of THE GUIDE by Mr. Govan. Mr. Lewis related to certain experiences with warehouses he had had, which were similar to those outlined by Mr. Govan, in which the heating plant was more than sufficient to take care of the requirements of the building.

R. C. Bolsinger, Philadelphia, asked if the same conditions existed when steam was used for heating the buildings as when hot water was used, to which Mr. Govan replied that he had had no experience on this subject with steam heating systems.

Champlain L. Riley, New York, stated that in his opinion the introduction in buildings of air cooling systems will force consideration of the factors of which Mr. Govan spoke. He thought that when refrigeration becomes more commonly used in the summer, the engineer will take the entire design in

hand and will provide buildings which will retard the passage of heat through the walls to the proper degree in the winter time, and hold the *cold* satisfactorily in the summer.

Mr. Pearce, Chicago, favored the idea, suggested by Mr. Govan, of taking the mystery out of the subject of insulation by changing the phraseology.

F. E. Giesecke, College Station, Texas, was of the opinion that the paper was a valuable contribution, and stated that it had given him an entirely new idea of the subject. S. R. Lewis called attention to the fact that statistics developed during the war indicated that about twice as much coal per square foot of radiation is burned in Texas as in northern Minnesota, indicating that as climatic conditions become more severe, the tendency is to increase the amount of heat resistance of buildings correspondingly.

The question of over-radiating the buildings described by Mr. Govan was brought up by L. A. Harding, who asked if the infiltration was calculated on the basis of leakage of periphery of the sash or on air change basis. Mr. Govan replied that on account of the fact that the windows used were of a special nature, and that consequently no infiltration data was given in THE GUIDE, the air change method was used. Mr. Govan further stated that the calculations had been checked by a member of the council, and that if anything, the infiltration requirements were underestimated.

Mr. Harding pointed out that it is quite possible to greatly overestimate the radiation requirements of a building by using the air change method instead of the infiltration per foot of sash method. To offset the suggestion that the apparent overradiating of the building was due to infiltration calculations, Mr. Govan stated that considerable difficulty was experienced in keeping the windows closed even during extremely cold weather. Mr. Harding asked Mr. Govan if in his opinion the difference in the actual and theoretical results was due to incorrect transmission coefficients in THE GUIDE, to which Mr. Govan replied that in his opinion THE GUIDE coefficients are correct, but that the assumption as to outside temperatures was in error on account of the fact that the heat capacity of the building will in most cases bridge over daily fluctuations in temperature.

The paper entitled Overall Transmission Coefficients Obtained by Test and by Calculation by Rowley, Algrin and Blackshaw, was submitted by Professor Rowley. (Paper published May JOURNAL, p. 49.) In a written discussion of this paper, H. S. Ashenhurst, Chicago, emphasized the importance of proper construction, and stated that in his opinion the Society could do no greater work for the small house owner than to make it its business to see that insulation is properly installed to secure maximum results. A written discussion was also submitted by C. K. Swift, Camden, N. J., who stated that this paper has definitely settled the long disputed question as to the accuracy of calculated heat transmission coefficients. Mr. Swift questioned the advisability of carrying calculations for heat transmission coefficients beyond significant figures.

In discussing this paper, F. C. Houghten stated that in his opinion it fills a very important gap in our understanding of heat transmission through building construction, because it proves conclusively that overall transmission coefficients can be computed with accuracy. This, he stated, is a very important factor when it is considered, that because of the infinite number of combinations of materials possible, transmission coefficients by test are not practical.

It was the opinion of Professor Kratz that one of the most important facts brought out by this paper is that the engineer's calculations can be nullified by improper construction, and that more attention must be paid to proper construction and supervision. Professor Kratz called attention to the possibility of using inside surface temperatures of walls as a means of evaluating them. Professor Rowley closed the discussion by stating that he hoped that the method of checking the walls after they are installed, suggested by Professor Kratz, could be developed.

The report of the Committee on the Code for Heating and Ventilating Garages (Published in May 1929 JOURNAL, p. 63) was submitted by W. H. Carrier, a member of the Committee, in the absence of E. K.

SOME OF THE ONTARIO CHAPTER COMMITTEE MEMBERS

TOP ROW: MRS. W. J. OLVANY, DEAN F. PAUL ANDERSON AND MRS. L. S. STILES. BOTTOM ROW: M. BARRY WATSON, TORONTO, GENERAL CHAIRMAN OF COMMITTEE ON CONVENTION ARRANGEMENTS; MRS. WATSON; MR. AND MRS. JOHN F. HALE, CHICAGO

Campbell, Kansas City, chairman, who was injured in an automobile accident enroute to the convention.

A written discussion of this report submitted by J. M. Frank, Chicago, suggested that phase protecting devices be furnished by the owner or wiring contractor and not by the motor or fan manufacturer. He also asked that the expression *non-sparking* be clarified. Mr. Frank stated that in the code the statement is not clear regarding the continuance of ventilation of pits by a system independent of the main garage ventilating system. Upon motion by W. T. Jones, the report was unanimously accepted and adopted as a code of the Society.

The following resolutions submitted by W. H. Driscoll, Chairman of the Resolutions Committee, were unanimously adopted:

Resolved, that the congratulations of this Society be extended to the members of the Ontario Chapter, for the thoroughness with which they have attended to the many details necessary to the carrying out of a meeting with such a high measure of success as has attended their efforts here; and be it further

Resolved, that the thanks of the Society be conveyed to the Ontario members for providing a meeting place so unique and so delightful, and for a program of entertainment and sports that has so happily provided for the pleasure of every visiting member and guest.

Resolved, that this Society express its appreciation to the charming women of Ontario who served so splendidly as members of the Ladies' Committee, and whose presence helped make this meeting the great success that it is; and that the Society express its regrets that Mrs. Arthur S. Lietch, the Convener of the Ladies' Committee should have been unable to be with us because of the sudden illness of her daughter, and express its hope for the speedy recovery of the young lady.

Resolved, that it is the sense of this meeting that the Ontario Chapter could have made no better selection for the place of this meeting than Bigwin Inn; and that we express our appreciation to the management and staff of the Inn, who, confronted with the unusual situation of having to open up in advance of the regular season, for our accommodation, have met our needs so skillfully and satisfactorily.

Resolved, that by popular acclamation this gathering place itself on record expressing its admiration for the amazing skill exhibited by the Skipper of the Toonerville Trolley in conveying such a large number of our members and guests in perfect safety over the hazardous and tortuous passage of the Portage.

Report of Exposition Committee

To the Council,
American Society of Heating & Ventilating Engrs.,

Your Committee on the International Heating and Ventilating Exposition wishes to report that under the able management of the International Exposition Co., plans for the International Heating and Ventilating Exposition are progressing rapidly, and this exposition should prove highly successful from every standpoint.

Definite arrangements have been made to hold the exposition in the Commercial Museum, in Philadelphia, from January 27 to 31, 1930, inclusive.

The Commercial Museum is admirably fitted for the holding of such an exposition, as all exhibits will be on one floor, which contains approximately 100,000 sq. ft. of space.

To date, definite contracts have been signed with 120 exhibitors and the space so taken represents approximately 40 per cent of the total space available.

With the exposition still six months in the future, it is believed that no difficulty will be experienced in contracting for space, and it is further believed that every phase of heating and ventilating industry will be represented at this exposition.

Great care has been exercised by your Committee and the International Exposition Co., to eliminate all companies whose products do not directly bear on heating and ventilating work.

The members of the Society can do much toward making this exposition a definite success, by urging heating and ventilating manufacturers to use space and exhibit their products.

Respectfully submitted by Advisory Committee of International Heating & Ventilating Exposition.
June 28, 1929: H. P. Gant, *Chairman*

After a few appropriate remarks by Chairman Lewis, the third and final session adjourned.

Entertainment and Sports at Bigwin

EVERY possible wish of the 380 people who attend the Society's Meeting at Bigwin Inn, had been anticipated by M. Barry Watson, General Chairman and his aids from the Ontario Chapter. Motorists had been furnished with full touring directions and at the entrance to the City of Toronto, Chapter representatives were stationed to direct the visitors along the posted streets to the King Edward Hotel, where registration headquarters and reception rooms had been established. For those who came to Toronto by train, Chapter representatives from M. F. Thomas' Reception Committee met them at the Union Station where a registration desk had been placed to give complete information about the remainder of the trip.

The travelers from various cities, coming by train, joined at Toronto, and at noon boarded a special train on the Canadian National Railways for the trip to Bigwin Inn via Huntsville. This special was an eight-car train with two dining cars in which a special luncheon was served.

On the way to Huntsville, the Ontario Chapter and hotel representatives passed out the special convention badges and assigned hotel rooms so that there would be no delay in registering and getting settled at Bigwin.

Upon arrival in Huntsville, a crowd of 120 boarded a lake steamer which brought them to the Portage where they enjoyed the experience of riding on a narrow gage railway to the Lake-of-Bays, where another boat brought them to the dock at Bigwin.

When this crowd arrived and joined those who

A Group of Members and Guests at Bigwin

had come by motor via Norway Point, the population at Bigwin Inn, had been increased by over 300. The visitors with appetites sharpened by their long ride made a rapid foray on the dining room located at the lake side.

After dinner the crowd assembled in the rotunda where M. Barry Watson, chairman of the Committee on Arrangements for the Ontario Chapter welcomed them to Canada and introduced A. S. Leitch, chairman of the Entertainment Committee who in turn presented the members of his Committee in charge of the various events such as golf, tennis, swimming, lawn bowling, etc.

Mrs. Thornton Lewis, wife of President of the Society and Mrs. M. Barry Watson, wife of the President of the Ontario Chapter were the recipients of baskets of beautiful flowers. Following the introduction of the committee members, everybody went to the pavilion where an informal dance was enjoyed.

On Wednesday during the technical session, the ladies walked around Bigwin Island and climbed to the observation tower located at the highest point on the island where they could get a splendid view of the surrounding country and the many coves and indentations which are characteristic of the Lake-of-Bays.

In the afternoon a steamer left Bigwin Inn dock and a three hour sail acquainted them with much of the beauty which makes the Lake-of-Bays so fascinating as a summer resort.

While the ladies were enjoying their lake trip, the men played a qualifying round in preparation for the Research Cup tournament which was held on Thursday afternoon. In this Tournament of Lost Balls, despite such hazards as water, forest lined fairways and mental handicaps some good scores were turned in. The weather was under good control so that members had two perfect days for golf.

On Thursday in the Research Cup tournament, L. J. Pitcher, Chicago, captured the cup for the second consecutive year. The golfers were divided into three classes, A and B with handicaps from 0-17, Class C 18-24 and Class D 25-30. Par for the course was 66 and the low net score in the A and B Class was made by L. J. Pitcher with John Howatt, runner-up. The low gross score was made by A. B. Darling, Montreal. In Class C, L. K. Berman, New York had the low gross score and H. G. Thomas, Chicago had the low net score, in Class D, C. V. Haynes, Philadelphia, had low gross score and A. B. Knight, Detroit, had low net score.

In the ladies golf tournament held on Thursday

Some of Members and Guests Who Attended Semi-Annual Meeting

morning, Mrs. F. A. Kitchen, Kansas City, had the low gross score, Mrs. Gurney, and Mrs. Beatty, both of Toronto, tied for low net score.

The Semi-Annual Banquet was held in the main dining room on Wednesday evening, June 26, and a special menu was served. During the courses, the Bigwin Inn orchestra entertained with musical and vocal selections.

M. Barry Watson acted as toastmaster and toasts were made to the King and to the United States. The entire assembly sang the first verse of the British National Anthem and the first verse of America.

A brief talk was made by Pres. Thornton Lewis of the Society and Vice-President L. A. Harding entertained the gathering with a few remarks on "research."

W. H. Driscoll presented the following resolution which was unanimously adopted:

WHEREAS: The American Society of Heating and Ventilating Engineers is assembled in its 35th Semi-Annual Meeting, and

WHEREAS: For the first time in its history, the Society is meeting outside the limits of the United States, and

WHEREAS: This most successful and delightful event is being held under the auspices of the Ontario Chapter and within the confines of the British Empire; therefore,

BE IT RESOLVED: That the greetings of this Society be extended to the members of the *Institution of Heating and Ventilating Engineers of Great Britain;* and

BE IT FURTHER RESOLVED: That as an additional mark of our esteem these Greetings be personally conveyed and delivered to the President of the *British Institution of Heating and Ventilating Engineers* by the President of this Society, Mr. Thornton Lewis.

The address of the evening was given by Brig. General C. H. Mitchell, dean of the Faculty of Applied Science and Engineering of the University of Toronto.

At the conclusion of General Mitchell's talk the members and guests enjoyed an hour of dancing in the pavilion.

One of the high points in the entertainment arranged for by the Ontario Chapter members was the theatrical entertainment and masquerade which was held in the pavilion on Thursday evening at 8:30. An "international" polo match between the Canadians and the Americans was greatly enjoyed. The dashing of steeds and the clash of sticks made it a most exciting event. A chariot race was greatly enjoyed and the introduction of the world's strongest man and others of unusual physical and mental attainments brought great applause from the audience. A grand march in the pavilion was held and those that were masked and wore unusual costumes passed before the judges and were awarded suitable prizes. The most original ladies costume was worn by Mrs. Wm. G. Boales of Detroit and the most original man's costume disclosed John F. Hale, of Chicago. The best costume for lady and gentleman were worn by Mrs. Holt Gurney of Toronto and M. Barry Watson of Toronto. Another costume of unusual merit was that of Dean F. Paul Anderson, Lexington, Ky., who portrayed George McManus' famous character, Jiggs.

In the tennis tournaments conducted under the direction of G. A. Playfair, the ladies singles tournament was won by Mrs. W. T. VanAllen, Pittsburgh and second place was taken by Mrs. A. A. Fitzgerald, Penetanguishene, Ont., while first place in the men's singles went to T. A. Worthington, Toronto, second place to L. E. Stiles, Brooklyn, N. Y. The mixed doubles event was won by Mr. and Mrs. T. J. Duffield, New York.

The best lawnbowlers in the ladies tournament were Mrs. J. J. MacKenzie, Toronto and Mrs. Warren Webster, Jr., Camden, N. J.

On Thursday afternoon the ladies enjoyed a bridge and tea on the balcony of the rotunda and there were 19 tables with a suitable prize for each table.

Other games and sports that were enjoyed were badminton, swimming, sailing and motor boating. M. C. Barnum, Newark, N. J. and H. F. Horton, Chicago, found sailing a pleasant pastime and S. R. Lewis, Chicago, took time enough from his official duties to handle a tiller. Mrs. L. A. Harding and Mrs. C. V. Haynes received some thrills in driving speedboats, while Warren Webster, Jr. and R. T. Coe discovered that there were fish in the Lake-of-Bays.

With the variety of things to do and the many

A Group of Members and Guests at Bigwin Inn

places to go, the hours not devoted to technical sessions and committee meetings were well filled.

The Semi-Annual Meeting 1929 was one of the largest ever held by the Society and the Ontario Chapter members and committees were greatly pleased that the members who attended, found the meeting so enjoyable.

Word of an unfortunate accident came in a message from Kalamazoo, Mich., telling of the injury of E. K. Campbell, Kansas City in an automobile crash while driving to Bigwin Inn. A message from the Society sent to him at the hospital expressed the sorrow of the members for his misfortune and hoped for his speedy recovery.

Because of the sudden illness of her daughter, a few days before the meeting started Mrs. A. S. Leitch, chairman of the Ladies Committee for the Ontario Chapter was prevented from attending

Due to the absence of A. J. Dickey in Europe the financial details of the meeting were handled by M. W. Shears.

TECHNICAL PROGRAM

Wednesday, June 26

8:30 a.m.—Registration.

9:30 a.m.—Greeting by Ontario Chapter President.

Response by Pres. Thornton Lewis.

Instruments for the Measurement of Air Velocity, by Prof. J. H. Parkin.

Analysis of the Over-all Efficiency of a Residence Heated by Warm Air, by Prof. A. P. Kratz and J. F. Quereau.

Air Conditioning System of a Detroit Office Building, by H. L. Walton and L. L. Smith.

Heat and Air Volume Output of Unit Heaters, by Prof. L. S. O'Bannon.

Report of Committee on Code for Testing and Rating Unit Heaters, by D. E. French, Chairman.

Thursday, June 27

9:30 a.m.—Errors in the Measurement of the Temperature of Flue Gases, by Percy Nicholls and W. E. Rice.

Pipe Sizes for Hot Water Heating Systems, by Prof. F. E. Giesecke and E. G. Smith.

Report of Guide Publication Committee, by S. R. Lewis, Chairman.

Capacity of Radiator Supply Branches for One and Two-Pipe Systems, by F. C. Houghten, M. E. O'Connell and Carl Gutberlet.

Report of Committees on Interpretation of Code for Rating Low Pressure Heating Boilers, by L. A. Harding, Chairman.

Friday, June 28

9:30 a.m.—Determining the Quantity of Dust in Air by Impingement, by Prof. F. B. Rowley and John Beal.

Five Suggested Methods of Appraising Insulations, by Paul D. Close.

Time Lag as a Factor in Heating Engineering Practice, by James Govan.

Over-all Heat Transmission Coefficients Obtained by Tests and by Calculation, by F. B. Rowley, A. B. Algren and J. L. Blackshaw.

Report of Committee on Code for Heating and Ventilating Garages, by E. K. Campbell, Chairman.

Report of Advisory Committee on Heating and Ventilating Exposition, by H. P. Gant, Chairman.

ENTERTAINMENT EVENTS

Tuesday, June 25

12:00 a.m.—Special train leaves Toronto for Bigwin Inn via Huntsville.

6:00 p.m.—Arrive Bigwin Inn—Dinner from 6:30 to 8:00 p.m.

8:15 p.m.—Special program for Ladies—Introduction of Chapter Officers in the Rotunda.

9:00 p.m.—Informal reception and dancing in the pavilion. Motion pictures in the Rotunda.

Wednesday, June 26

8:30 a.m.—Registration in the pavilion.

10:30 a.m.—Special hiking party for ladies to explore Bigwin Island.

1:30 p.m.—Special Boat trip around Bigwin Island and Lake-of-Bays for ladies.

2:30 p.m.—Qualifying Golf Match.

7:00 p.m.—Semi-Annual Banquet in main dining room followed by dancing.

Thursday, June 27

9:30 a.m.—Golf tournament for ladies.

2:30 p.m.—Bridge party and tea for ladies.

2:30 p.m.—Golf tournament for Research Cup.

8:30 p.m.—Amateur Theatricals by cast of Ontario Chapter's members followed by Masquerade Ball in the Ballroom.

Friday, June 28

2:00 p.m.—Golf Match.

Tennis Tournament.

Lawn Bowling.

International Bang-and-Go-Back Motor Boat Racing Contest (each Chapter will have a boat).

Log Rolling Contest by local talent.

COMMITTEE ON ARRANGEMENTS

M. Barry Watson, *General Chairman*
M. F. Thomas, *Reception*
A. J. Dickey, *Finance*
Arthur S. Leitch, *Entertainment*
E. B. Sheffield, *Transportation*
E. M. Dolan, *Publicity*

Ladies Committee on Arrangements

Mrs. A. S. Leitch, *Convener*

Mrs M. F. Thomas
Mrs. M. B. Watson
Mrs. E. B. Sheffield
Mrs. A. J. Dickey
Mrs. H. R. Flett
Mrs. R. W. M. McHenry
Mrs. H. H. Angus

Officers of Ontario Chapter

M. Barry Watson, *President*
H. J. Church, *Vice-President*
J. S. Paterson, *Secretary*

Board of Governors

M. Barry Watson
R. W. M. McHenry
H. J. Church
E. B. Sheffield
J. S. Paterson
A. J. Dickey

CANDIDATES FOR MEMBERSHIP

The Constitution of the Society, as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the name of applicants and their references shall be printed in the next issue of the Journal of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Membership Committee as soon as possible.

When the Membership Committee has acted favorably upon a Candidate's application and assigned his grade the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 17 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Membership Committee, and in turn the Council, urge the members to assume their share of the responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned. All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some member by September 5, 1929, these candidates will be balloted upon by the Council. Those elected to membership will be notified by the Secretary, immediately after election.

REFERENCES	Proposers	*Seconders*
DAVIS, OTIS E., Sales Engr., Hoffman Specialty Co., Scottsbluff, Nebr. (*Advancement*)	F. H. Gaylord F. H. Willis	Chas. W. Adams Wm. McCoy Larimer
GROSECLOSE, JOHN B., Dist. Mgr., Warren Webster & Co., 412 Builders Exchange Bldg., San Antonio, Texas.	M. L. Diver Wm. A. Ebert	Reginald F. Taylor
GUNTHER, RAYMOND CHESTER, Sales Engr., Carrier Engrg. Corp., Land & Title Bldg., Philadelphia, Pa.	Walter A. Bornemann P. L. Davidson	W. G. Hillen W. H. Carrier
JOHNSTON, ROBERT ELLIOTT, Cons. Engr., 3342 33rd Ave., W., Vancouver, B. C. (*Advancement*)	H. J. McCreery A. H. Blake	
KILNER, JOHN SAUNDERS, Partner, Kilner-Mills Co., 3-266 General Motors Bldg., Detroit, Mich.	J. H. Walker H. L. Walton T. E. Coon	J. E. Degan Geo. H. Giguere F. W. Johnson
KUEMPEL, LEON L., Sales Engr., Minneapolis-Honeywell Regulator Co., 2920 Fourth Ave. S., Minneapolis, Minn.	John V. Martenis Frank B. Rowley	H. J. Sperzel Milton Wunderlich
LAWSON, WILLIAM IRVIN, Vice Pres. & Gen. Mgr., The Buckeye Blower Co., Columbus, O.	Dr. E. V. Hill M. F. Rather F. C. McIntosh	H. W. Page A. J. Nesbitt J. H. Kitchen
MALONE, DAYLE G., 7015 Merrill Ave., Chicago, Ill. (*Advancement.*)	J. C. Bostain Wm. J. Doyle	A. C. Willard E. J. Gossett
METCALF, RALPH H., C. A. Dunham Co., 3605 Laclede Ave., St. Louis, Mo.	M. D. Pence Macy S. Good	C. A. Dunham C. A. Thinn
MOFFITT, ROY MICHAEL, Br. Mgr., L. J. Mueller Furnace Co., 200 Reed St., Milwaukee, Wis.	E. A. Jones C. W. Miller	H. K. Lees W. B. Graves
PHILLIPS, WILLIAM J., 520 W. First St., Royal Oak, Mich.	D. H. Petherick R. S. M. Wilde	R. K. Milward J. F. McIntire
SELCH, HOMER CORNELIUS, Heating Contr., 844½ Virginia Ave., Indianapolis, Ind.	G. A. Voorhees Geo. O. Jackson	H. W. Neal R. H. Edwards
SHAVER, HERBERT H., 215 Wheeler Ave., Scranton, Pa.	R. V. Frost F. D. Mensing	M. C. W. Tomlinson E. N. Sanbern
STITT, HOWARD BENJAMIN, National Radiator Corp., 431 W. Georgia St., Indianapolis, Ind. (*Advancement.*)	C. C. Shipp George O. Jackson O. S. Hill	C. E. Gronberg H. L. Repp I. G. Belzor
TRUE, JOHN E., Dist. Mgr., York Htg. & Vtg. Corp., 622 Broadway, Cincinnati, O.	Thornton Lewis Donald E. French	Jas. V. Cavileer H. P. Gant
WALKER, GEORGE EDWARD, Sales Engr., Weil Pump Co., 215 W. Superior St., Chicago, Ill.	R. O. Nelson Jean S. Jenson	C. E. Crone, Jr. Geo. E. Wells (non-member)
WILLIAMS, DAVID BUTTLER, Constr. Draftsman, Carrier Engrg. Corp., 1724 Land Title Bldg., Philadelphia, Pa.	Walter A. Bornemann P. L. Davidson	W. G. Hillen L. L. Lewis

Candidates Elected August, 1929

In the past issues of the Journal of the Society there were posted among the list of Candidates for Membership 44 candidates for election in the different grades of Membership, whose names have been balloted upon by the Council. We are now instructed by the Council to post herewith, as required by Art. II, Sec. VIII, of the By-Laws, the following list of candidates elected:

MEMBERS

AHEARN, WILLIAM JOSEPH, Htg. Engr., The Downey Co., 99 W. Dedham St., Boston, Mass.

DINGLEMAN, CHARLES S., Chief Engineer, Stanley Company of America, 1132 Fidelity-Philadelphia Trust Building, Philadelphia, Pa.

ELLIOT, EDWIN, Sales Engr., Edwin Elliot & Co., 560 North 16th St., Philadelphia, Pa.

FLEMINGS, JOHN A., Mgr., Spencer Heater Co., 216 Tremont St., Boston, Mass.

HIERS, CHARLES R., U. S. Radiator Corp., 101 Park Ave., New York, N. Y.

JOHNSON, WALTER F., Mgr., American Radiator Co., 374 Delaware Ave., Buffalo, N. Y.

LELAND, WARREN B., Sales Engineer, The H. B. Smith Company, Westfield, Mass.

MEHNE, CARL A., Engr., Richardson & Boynton Co., 260 Fifth Ave., New York, N. Y.

NAROWETZ, LOUIS LUDWIG, Jr., Secretary and Active Head, Narowetz Htg. & Vtg. Co., 1711 Maypole Ave., Chicago, Ill. (*Advancement.*)

NASON, GEORGE L., Buerkel & Co., Inc., Boston, Mass.

O'NEILL, JAS. WALTER, Chief Engineer, The Trane Company of Canada, 439 King W., Toronto, Canada.

RIES, LESTER, Supt. of Bldgs., University of Chicago, 5750 Ellis Ave., Chicago, Ill.

SANDS, CLIVE CHISHOLM, Asst. Mech. Engr., Federal Capital Commission, Canberra, F. C. T., Australia.

SHANKLIN, ARTHUR P., Sales Engr., Carrier Engrg. Corp., 850 Frelinghuysen Ave., Newark, N. J.

STEEN, JOSEPH M., Iron City Heating Co., 843 Jacksonia St., N. S., Pittsburgh, Pa.

SWAN, ERNEST H., Senior Partner, W. A. Swan & Sons, 122 Unley Rd., Unley, South Australia, Australia.

WALKER, GEORGE E., Sales Engineer, Weil Pump Co., 215 W. Superior St., Chicago, Ill.

WASHINGTON, LAWRENCE W., Vice-Pres. and Sec'y, National Regulator Co., 2301 Knox Ave., Chicago, Ill.

WHITE, CHARLES FERBER, Sales Engr., Modine Mfg. Co., 3010 Market St., Philadelphia, Pa.

ASSOCIATES

ALCOTT, WILLIAM L., The Graff Co., 945 Liberty Ave., Pittsburgh, Pa.

CARLSON, EVERETT E., Branch Manager, The Powers Regulator Co., 1010 Louderman Bldg., St. Louis, Mo.

DAVIS, HERBERT H., Secy.-Treas., Herbert H. Davis Co., Inc., 4146 S. Western Ave., Chicago, Ill.

FEELY, FRANK J., Mgr. of Sales, The Taylor Supply Co., Detroit, Mich.

HOOD, LESLIE A., Sales Engineer, Trane Company of Canada, 439 King St., W., Toronto, Canada.

MILLER, TOLBERT G., Supt., Herre Bros., 7 and Emeralds, Harrisburg, Pa.

MURPHY, JOSEPH R., Sales Manager, Taco Products, 342 Madison Ave., New York, N. Y. (*Advancement.*)

ROBERTS, CHARLES A., Resident Manager, Imperial Iron Corporation, Ltd., Toronto, Ont., Can.

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Heating · Piping
and Air Conditioning

EDITORIALS

Air Conditioning Growing Up

Probably every science had in its early days its period of twilight. Sometimes a science struggled along for centuries without receiving public recognition. It was felt, however, when a science was recognized by great educational institutions and was taught in them, that science then had passed the early morning hours of its life and was in the full light of the forenoon sun.

As an indication then of the progress of air conditioning, it is interesting to note that a number of American schools and universities are training engineers to become expert in this science. Among these are Harvard, Yale, Columbia, Michigan, Ohio State, Kentucky, Illinois, Purdue and Massachusetts Institute of Technology. These schools and universities are specializing in air conditioning and there has been a tremendous increase in the number of students in these sciences.

In addition to this, various studies which pertain to the science of air conditioning are taken up at Carnegie Institute of Technology, Pittsburgh, for instance, where recently there has been started a short course on fans which is available to students. Mellon Institute recently completed extensive studies of the soot and smoke problem and methods of combating it.

Air conditioning, the youngster of a decade ago, is growing up.

In Step With Progress

Where the sciences of air conditioning, piping and heating stand mechanically, is to be revealed in an exposition of products of these industries to be held in the winter of 1930. This exposition will be conducted in connection with the 1930 annual meeting of the American Society of Heating and Ventilating Engineers in Philadelphia. It promises to enlighten air conditioning, piping and heating engineers and the layman as well.

It is difficult to realize the increased momentum that has been given to the world by means of the products of the designing and manufacturing engineer. His influence is seen in almost every country in the world. Civilization is adjusted to a new tempo through the product of his study and skill. What man more than he has contributed to the advancement of the world?

The product of the designing and manufacturing engineer reduces hazards, eliminates nuisances, adds comforts heretofore undreamed of, and saves man's most valuable commodity—time.

We are astonished at the high steam pressures, 1800 pounds or more per square inch. But what of the future? Professor Bridgman of Harvard has, in another field of engineering, succeeded in exerting a pressure of 600,000 pounds per square inch. It is only the engineer with the imagination of genius who can look in to the future and anticipate what these stupendous pressures may mean.

This so-called sordid age of metals and machinery is, in the light of these developments, no longer sordid. It is filled with romance, the zeal for and thrill of achievement beyond that in any of the story books of old, and has in it the potentialities of human comforts which the world today can not even imagine.

The engineer who refuses to learn from such expositions the trend of affairs in the sciences of piping, heating and air conditioning, is standing still and this, because of the progressive movement around him, means putting a greater distance between himself and the front ranks.

Formulas, Charts, Tables vs. Guessing

"Why do you publish so many tables, charts, formulas, etc.?" inquires a reader of this paper. "We want practical, everyday information." Yet the next two letters are likely to tell us that our information is practical and usable. Apparently different readers approach their technical reading in different ways.

The time was when any temperature between 55 and 90 in the old log house was considered fairly satisfactory. There was a time when fifteen miles an hour behind a smoke-belching locomotive filled our highest hopes of rapid travel. But things have changed and the changes have come about through the accuracy of engineering knowledge. But accurate data must be set down so it can be studied and applied and so we have turned to the use of charts, formulas, tables, etc.

A writer may use words, phrases and sentences to describe a majestic building but these would not do for the engineer. He must have the specific facts as revealed by charts, tables, formulas. They may

show the difference between the success and failure of a costly piece of equipment. They may point the road to 10 per cent profit instead of 10 per cent loss. They may reveal the unexpected in the physiological reactions to atmospheric environment and so enable the fabricator of engineering equipment to protect the lives and health of the people.

Charts, tables and formulas are not as formidable as they seem. In fact, there is nothing in engineering that is not simple when it is learned. The pseudo engineer who will take the time and trouble to study the tables, charts and formulas in this paper and learn how to adapt them to his uses, will be taking a step toward accuracy and a step away from the guess-so's of former years.

Heating, Ventilating and Air Conditioning

Heating for comfort has, in the later years, taken on a somewhat definite meaning. It means heating to 70 degrees or thereabouts. In the old log house with its fireplace it might be 90 degrees near the fireplace and 50 degrees at the opposite end of the room. No one at that time found fault with such wide variations. But no one today would buy a heating plant that would not function any better than that.

The same reasoning may be applied to ventilation and that more modern development called air conditioning. Just what is the difference?

There exists, of course, some relation between heating—ventilation and air conditioning—yet heating, considered alone, may be accomplished in many different ways without regard to either of the others, ventilation or air conditioning. So also may ventilation be produced without considering or resorting to heating or air conditioning. Air conditioning, on the other hand, although perhaps erroneously considered by some few to be a glorified name applied to ventilation, is in reality much more. It embraces the control of any predetermined temperature and humidity conditions desired of the air in which we live, also the purity of the air and the air motion within a room, enclosure or entire building, regardless of the season of the year. Air motion is an important factor. Air motion and air distribution must be correct in order to permit uniformity of temperature and humidity conditions. Air is the means of conveying the proper amount of heating or cooling effect from the air conditioning apparatus to the room or rooms controlled.

Let us see how this differs from ventilation, as ventilation has been generally defined.

A fan, regardless of type, exhausting air from an enclosure produces ventilation by displacement, termed "exhaust ventilation." Turn this fan around and deliver outside air into the room and it is said to be supply ventilation. Heaters may be added to heat the air before delivery to the room and we then have a fan system of heating and ventilation. Add an air washer and we have a washed air ventilation and fan system of heating.

It is obvious that certain limitations exist in any of the above combinations of ventilation, mainly since no predetermined temperature and humidity conditions may be maintained except possibly during the winter months or heating season. Let us assume the following illustration and compare an office building equipped with a ventilation system and one equipped with an air conditioning system.

The building with a ventilation system will experience inside temperatures depending upon outside temperature conditions and the inside temperatures will vary with the outside temperature changes. The building cools down over night from the natural influences of the outside air cooling. As the outside air warms up the next day (it may reach 90 degrees) the ventilating system draws in the outside air and supplies it to the ventilated portion of the building. Obviously, the more 90 degree outside air that is brought into the building the higher will be the temperature of the air inside. A relative comfort is, of course, produced mainly by air motion and the displacement of inside air.

Considering the effect upon the inside temperatures only, it is generally understood that cooler temperatures would result if the windows were kept closed, shades drawn, and if no outside ventilated air was delivered to the building. This is reasonable. It is due to the stored up cooling effect of the walls, floors, furniture and equipment and lag in the temperature rise of the inside air as outside temperature changes, but in spite of the lower temperatures inside, the results would not be beneficial. The result would be a stuffy and close condition. How long these rooms would remain cool due to the natural lag would be dependent upon the number of people in the rooms, upon lights, etc.

Now take a building supplied with air conditioning. When the outside air rises to 90 degrees the inside conditions would be maintained at an even level, whether that is predetermined as 70 degrees, 80 degrees or any other temperature. It is by maintaining a uniform temperature and humidity condition, together with air motion, that one gives the greatest feeling of comfort to the occupants. It is something definite.

Just as heating has come to mean heating to 70 degrees in winter instead of anywhere between 50 and 90 degrees, so air conditioning has come to mean something definite. Temperature, of course, is only one factor in air conditioning. Air conditioning engineers believe it is just as necessary to maintain a definite humidity, definite purity and definite air motion in order to maintain maximum comfort.

And just as heating now maintains a definite standard, so air conditioning maintains a definite standard and is as different from ventilation as the old time fireplace is different from the modern up-to-date heating plant.

PRACTICAL PIPING PROBLEMS

Strainer for Oil and Water Lines
Water Supply Problems
Back-feeding Between Two Reservoirs

by
W. H. WILSON

Strainer for Fuel Oil or Water Lines

THE accompanying sketch shows a strainer that is made from standard pipe fittings and would be suitable for use on fuel oil lines.

The one shown was made by using two 12 x 12 x 4 standard flanged tees and was used on a fuel oil line straining the fuel oil as it was unloaded from oil tank railroad cars.

The assembly shows two baskets. The smaller or inner basket has a coarse mesh for collecting the larger particles of foreign matter that are frequently found in the fuel oil. The larger or outer basket has a finer mesh and the oil is given a double straining when this is used. The strainer baskets can be removed for cleaning by shutting down the oil line and taking off the top flanged cover.

The inlet and the outlet are shown opposite each other but may be faced at any angle to each other that the drilling template of the bolt holes in the flanges will allow.

The drain connection may be tapped in the side of the lower tee near the bottom if more convenient for installation.

As these strainers are used near the oil unloading tracks, it is customary to put them partly in the ground, drain being led off to a pit.

Referring to the assembly details, No. 4 is the inner basket. No. 1 is the top flange of the inner basket, the outside diameter of which is ¼ inch smaller than the inside bore of fitting No. 7, allowing a clearance of approximately ⅛ of an inch on each side to pass down through the body of the upper tee.

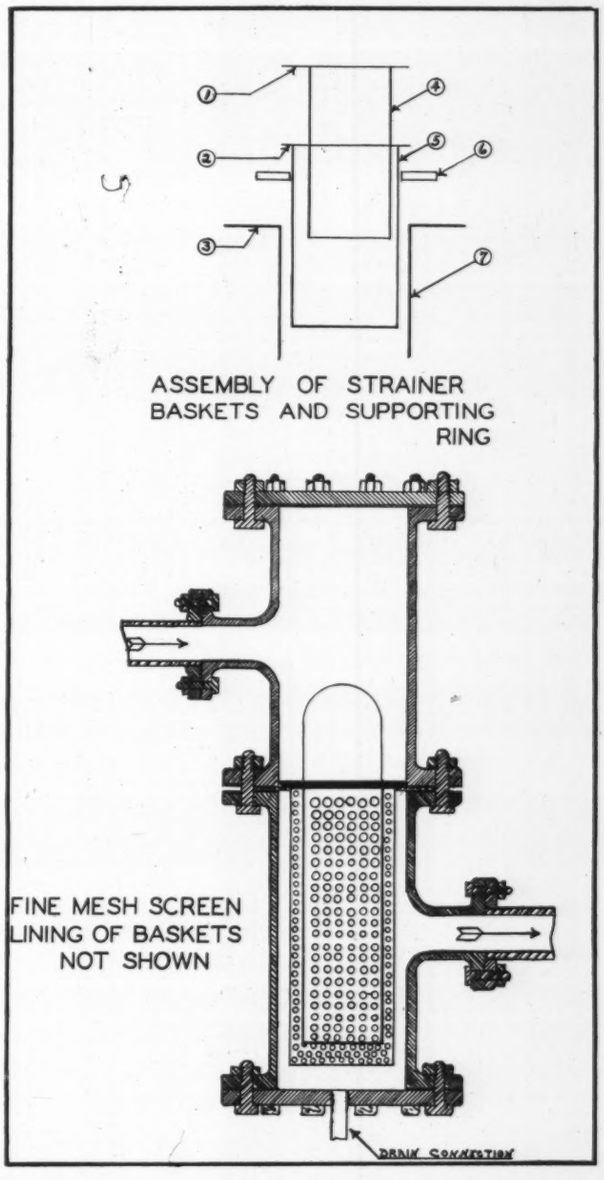
The outside diameter of the inner basket, No. 4, is approximately 2 inches smaller than the inside diameter of the outer basket, leaving approximately 1-in. clearance on each side.

The supporting ring No. 6 is made of steel, usually about ⅜-in. thick. The outside diameter of this supporting ring is the same as a regular ring gasket, which fits inside the bolt holes of the flanged tees. The inside diameter of the supporting ring is slightly larger than the outside diameter of the outer basket, permitting the basket to pass down through the opening in the ring until the top flange of this baskets rests on it.

The outside diameter of the top flange No. 2 of the outer basket is the same as the diameter of the top flange No. 1 of the inner basket.

The inside diameter of the top flange No. 1 is slightly larger than the outside diameter of the inner basket, permitting the inner basket to pass down in the outer basket until it rests on the top flange of the outer basket.

The supporting ring supports the two baskets by the flange of the outer basket resting on the ring and the flange of the inner basket resting on the flange of the outer basket. A bail may be put on the baskets so that they may be easily withdrawn for purpose of cleaning.

The oil passes down through the baskets as shown. The baskets can be made of perforated sheet steel or brass for oil—brass if used on water. The size of mesh depends on the kind of oil or fluid to be handled.

Ordinarily the inner basket can be made of per-